# AN ANALYSIS OF OCEAN THERMAL ENERGY CONVERSION SYSTEMS

William Joseph Keller

# AN ANALYSIS OF OCEAN THERMAL ENERGY CONVERSION SYSTEMS

by

WILLIAM JOSEPH KELLER, JR.

B.S., United States Naval Academy
(1970)

SUBMITTED IN PARTIAL FULFILLMENT

OF THE REQUIREMENTS FOR THE

DEGREE OF MASTER OF SCIENCE

IN CIVIL ENGINEERING

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

June, 1976

## ABSTRACT

AN ANALYSIS OF OCEAN THERMAL ENERGY SYSTEMS

by

WILLIAM JOSEPH KELLER, JR.

Submitted to the Department of Civil Engineering on 7 May 1976 in partial fulfillment of the requirements for the degree of Master of Science in Civil Engineering.

The Arab oil embargo of 1973 brought into full focus the problem of reliance upon a single source of energy. In order to diffuse this reliance, other alternative sources of energy have been investigated with intensive interest in recent years. One of these alternatives studied has been ocean thermal energy, a form of solar energy.

Ocean thermal energy is a vast resource of potentially useable energy convertable to such forms as electricity through power generation. Research has been directed in this area in order to demonstrate feasibility, both technical and economic, of the concept. What has emerged are proposals for the construction of large monolithic concrete sea platforms to support a conversion plant.

The design and construction of these offshore platforms represents a challenge to engineers and a proper perspective to design and development is necessary. Strategies developed, experiences gained in the North Sea oil explorations, prefabricated bridges and shipbuilding provide an ample basis for consideration in design and construction.

Thesis Supervisor: J. J. Connor
Title: Professor of Civil Engineering

## ACKNOWLEDGEMENTS

I would like to thank Betty, my friend and wife, for her understanding, patience, and encouragement during my graduate study. Without her support and advice, ["Stop complaining, and get on with it!"] it would have been far more difficult and less rewarding.

The advice and assistance given to me by Dr. Robert Cohen, ERDA; Dr. Eugene Silva, NavFac; Dr. Loyd Louis, ChesDiv; and Dr. Gerhard Jirka, MIT have been most valuable and crucial to my effort.

The development of this thesis has been most rewarding as an educational experience and the assistance and counsel given me by Dr. J.J. Connor, Sc.D., Professor of Civil Engineering, my supervisor, were responsible for this. His interest, insights, and time are highly valued by this grateful student.

Finally, to Jennifer and Christian, for giving up time with their daddy so he could work and for giving me so much joy during my study.

TABLE OF CONTENTS

## LISTING OF ILLUSTRATIONS

LISTING OF TABLES

# CHAPTER 1

# THE OCEAN THERMAL ENERGY CONVERSION CONCEPT

## INTRODUCTION

The oil embargo in 1973 awoke the ordinary citizen to what some researchers had concluded years before. It brought home the stark reality that the American society is at a threshold in the methods of its utilization of energy resources for power generation. The age of cheap and abundant petroleum based fuel was drawing to a close and new sources of energy must now be sought if the projected national energy needs for both short and long term growth are to be satisfied. With the sharply increasing oil prices came renewed government interest in research and development efforts of alternate methods of energy utilization and power generation.

As the cost of power generation spiraled upward, the economies of more exotic types of energy sources became attractive enough to warrant serious consideration. Along with the more traditional method of power generation such as nuclear fission and coal-burning, such methods as nuclear fusion, geothermal, solar radiation, indirect solar, wind, tidal, and ocean waves have received serious consideration as alternatives. While no one assumes that neither any of these sources in the short term will provide cheap energy nor will any of them in the long run provide the panacea for the energy problem, they do have the potential of providing a balance to

the present sources of energy and will therefore assist in insulating the economy from the effects of an abrupt loss of one source of energy.

In what follows we will look at one of the more exotic and presently least competitive of the energy source, ocean thermal energy conversion, a form of solar energy. We will discuss the process from its inception in an historical context and then will discuss broadly some of the particular design considerations required for the theory to be applied in a power plant design.

## THE SEA AS A SOLAR ENERGY COLLECTOR

An enormous amount of energy beyond the more expansive needs of the industrial world is daily transferred upon the earth's surfaces through solar radiation. This energy is absorbed by these surfaces resulting in an increased temperature at their upper levels. Since the oceans comprise the greatest portion of the earth's surface, the majority of this solar energy is collected in the oceans--they are the largest solar energy collectors.

For this energy to be utilized, a method must be developed which can tap this collected energy and deliver it in a useable form. This is a difficult task as the enormity of the ocean collectors causes the energy to be diffused across vast areas of ocean surface. Its concentration is insignificant compared to conventionally used sources of energy such as combustion or

fission. However, if it were not for other phenomena occurring continually within the oceans, any method of collection proposed would involve the design of an apparatus of incomprehensible magnitude. It would be impossible to use this resource without such a massive drain of other resources that the entire undertaking would be considered economic folly.

But the oceans are not stagnant pools of water. They are continuously moving and circulating masses. The waters in the tropics which receive the most solar thermal energy flow towards the polar regions. As they flow to the higher latitudes, they gradually transfer heat to other sinks, i.e., wind masses and coastal areas, and cool. This colder water, being denser, slowly sinks and flows back toward the equatorial regions beneath the warmer water. These two counter streams reinforce existing temperature differentials occuring in sharply defined gradients, thermoclines, with the greatest consistent temperature differences occurring in the tropical regions. Figure 1 demonstrates a typical temperature profile for a tropical sea. This phenomina exists also in the Gulf Stream as it exits the Carribean.

From thermodynamics it is known that the existance of thermal differences is a necessary condition for ability to do work. In the tropical oceans there exists a 40°F differential within 1,000 feet of depth of the surface. Therefore, with the development of an apparatus capable of bringing these waters of different temperature together in a heat engine, a possible method for using the ocean's solar energy is available.

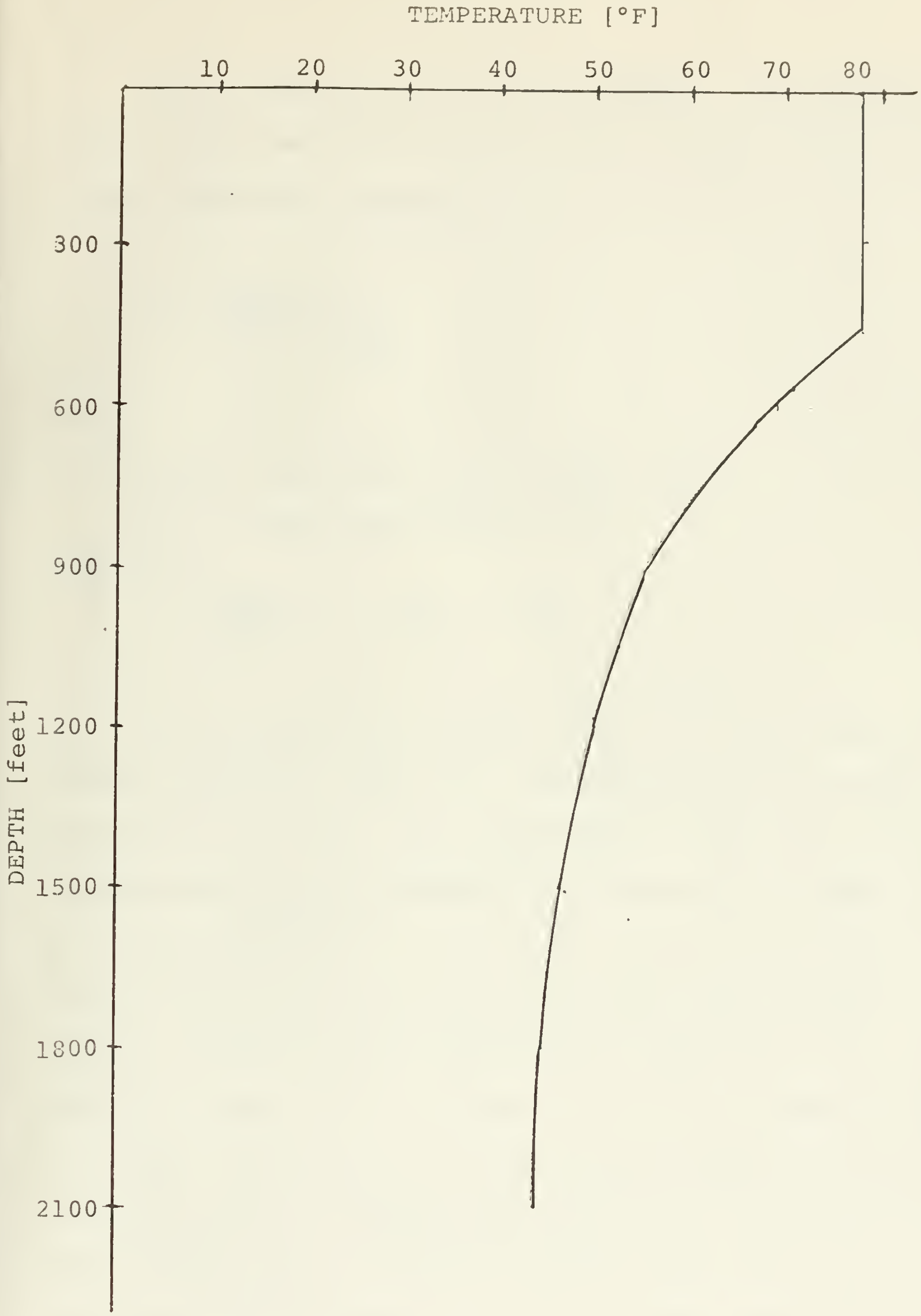



FIGURE 1

TYPICAL TROPICAL THERMOCLINE DIAGRAM

Due to the low $\Delta T$, 37°F for example, the efficiency of this energy conversion process will be low as demonstrated by Carnot efficiency equation:

$$\eta = \frac{T_{COLD}}{530°F + T_{HOT}}$$

when

$T_{COLD}$ = cold water inlet temp = 40°F

$T_{HOT}$ = warm water temp = 77°F

therefore

$$\eta = \frac{40}{530+77} \approx .06 \rightarrow 6\%$$

This 6% represents the optimal cycle efficient which after energy losses are taken into consideration will reduce to the range of 2.3% total cycle efficiency. While this efficiency is incredibly low as compared to conventional power plants with the efficiency of 10 to 40%, the available amount of energy is so large that even with such low efficiency a large amount of power can be generated if high flow rates of warm and cold water can be achieved. The Gulf Stream, alone, possesses the equivalent of 85 million megawatt hours of energy per year or seventy five times the projected U.S. energy requirements of 2.3 million megawatt hours per year in 1980. An Ocean Thermal Energy Conversion (OTEC) power plant to extract 100 megawatts of this energy will require, with a condensing temperature of 40°F, a total water flow rate of

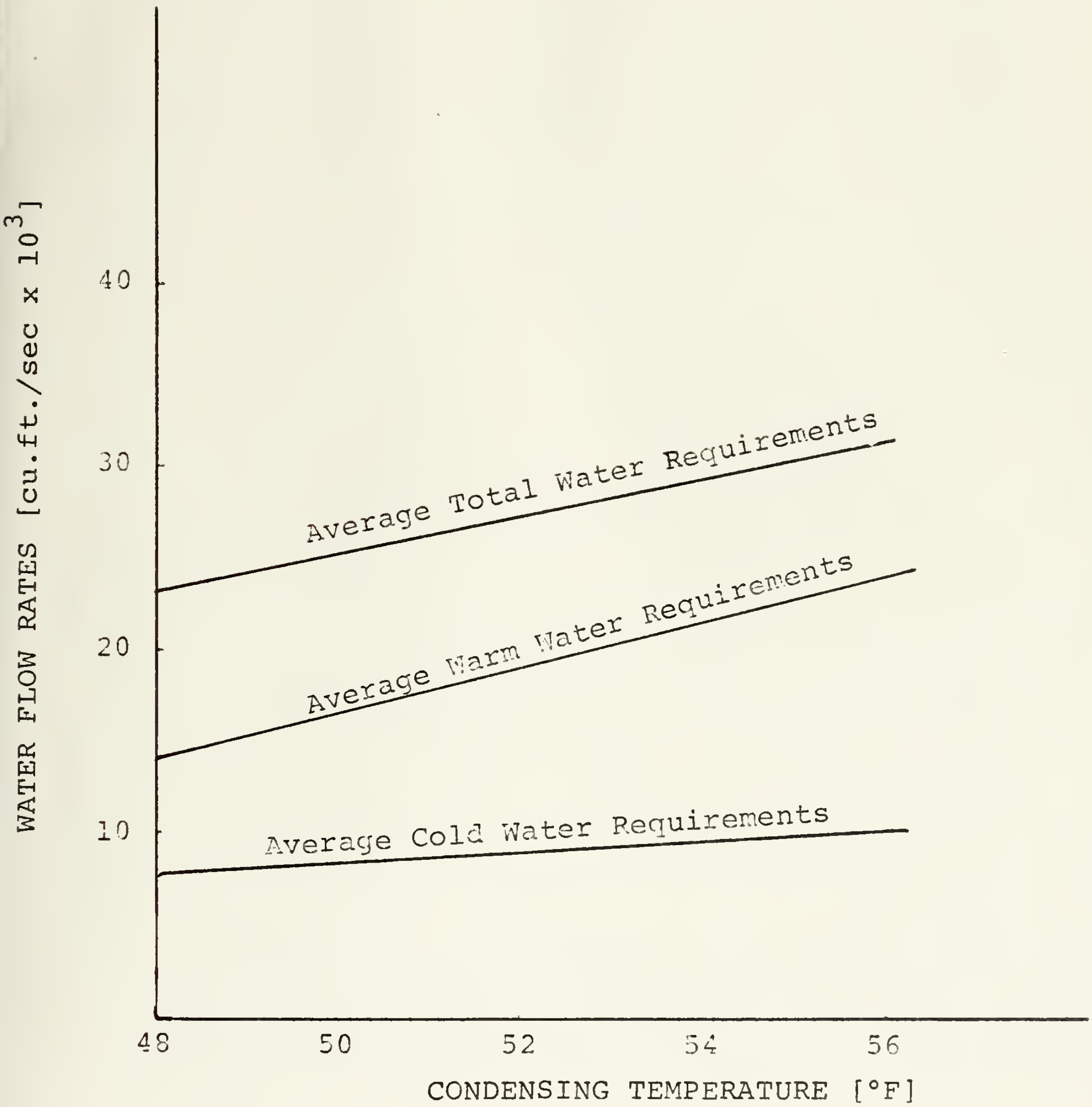



FIGURE 2

AVERAGE OTEC WATER FLOW RATES (1)

20,000 cubic feet per second. This type of flow rate is normally achieved in hydroelectric plants. Figure 2 illustrates required flow rate for a 100 $Mw_e$ OTEC plant versus condensing temperature.

## EVOLUTION OF THE CONCEPT

The concept of converting the ocean's solar energy into electrical power by utilizing the thermal gradient is almost 100 years old. It was first suggested by a French physicist, Jacques D'Arsonval, in Review Scientific on 17 September 1881. In this article, D'Arsonval hypothesized that through a heat transfer process, a working fluid, he suggested ammonia, could be used to drive a turbine-generator system producing a useable form of energy. This could be electrical energy.

No experimental results were to test the soundness of this hypotheses until 40 years later when another Frenchman, Georges Claude, constructed a 22 kilowatt net output plant on the shore of Matanzas Bay, Cuba, taking advantage of the area's topography with cold water at a depth of 100 to 200 meters within 200-300 meters of the shoreline and warm tropical surface water easily accessible. Although D'Arsonval had suggested using a working fluid in what will later be described as a closed Rankine cycle, the Claude plant employed an "open cycle" heat engine. (Figure 3) In this process, the warm water vapor with a flow rate of 500 meters per second was used to drive a large turbine. Upon being exhausted from the turbine, the vapors were condensed and were either pumped back into the ocean or, being fresh water, were retained for consumption.

Located on the shoreline, the Claude plant pumped the warm surface sea water into an evaporator under a vacuum. The pressure drop within the evaporator caused the seawater to

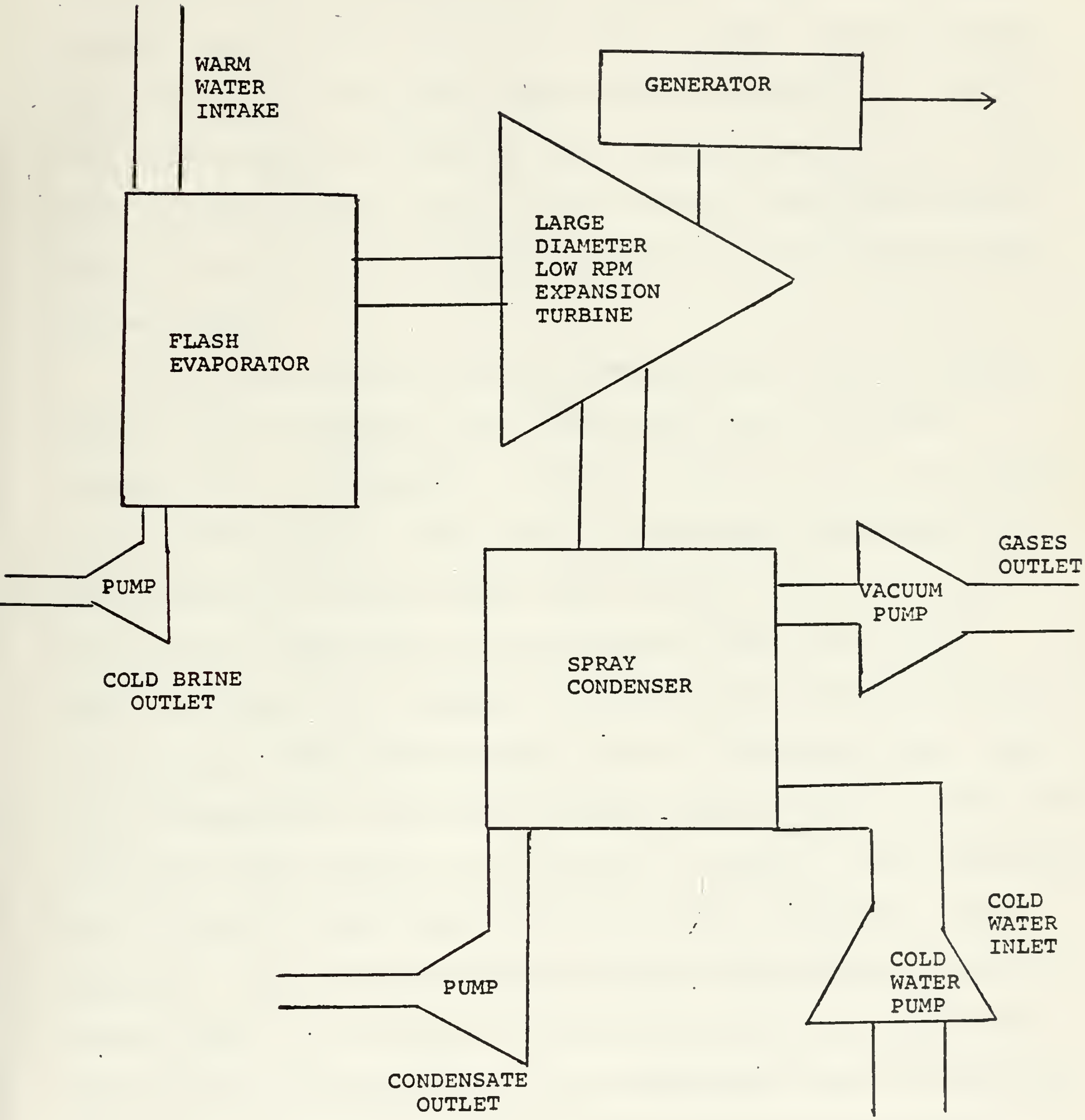



FIGURE 3

CLAUDE'S OPEN CYCLE PLANT(19)

form a saturated vapor and a brine, with the brine being pumped back out to sea. Deaeriation of this vapor which contained dissolved combustible gas was accomplished before the gas was allowed to expand in the turbine. Once through the turbine, the exhausted vapors were condensed in a shell-tube type condenser cooled by cold seawater brought up to the plant of one kilometer via a two kilometer corrigated steel pipe. This plant was the first operating system using ocean thermal energy.

As is the case with initial engineering applications of revolutionary concepts, the Claude plant was a failure economically. Its capital costs per kilowatt net output were prohibitively high. This high cost resulted from many problems inherent with the open cycle system. Since it used low pressure steam with a low Reynolds number and large specific volume, the turbines required were very large and inefficient. The direct vapor flows of the open cycle, even after deaeration, seriously corroded the turbine. Driving the expensive pumps required to maintain a vacuum in the evaporator and to draw the cold water into the plant, created a large parasitic loss upon the turbine output--approximately 50% of gross output. The length of the cold water pipe caused the cold water to absorbe heat as it transversed the tube with 16% (2°C) loss of available $\Delta T$ (11-13°C) lowering efficiency. This plant was both inefficient and expensive.

The Claude open cycle system was attempted later by the French in the 1950's in Abejan, Ivory Coast. Again taking advantage of local geography, they situated a plant on a shoreline in the immediate proximity of tropical waters underlied by cold artic waters. The ocean $\Delta T$ was 36°F. Water was brought up from a 3-mile depth via an 8" diameter pipe. The size of the plant, 3.5 Mw, was more respectible to demonstrate feasibility. The French made a serious effort on this project, yet the problems which plagued the Claude plant remained. The turbines were still large and inefficient; corrosion plagued the cold water pipe; and parasitic power losses devastated the net power output. (Loss of 50%)

Although some work continued and still continues on the development of an economical open cycle, the latest work conducted by the University of Massachusetts at Amherst again demonstrated the two inherent problems of the open cycle, high parasitic power losses and large uneconomical turbines. Table #1 shows the results of the UMASS open cycle design study and summarizes the parasitic losses of this design. There is some hope that the parasitic power loss problems can be partially alleviated through such techniques as controlled flash evaporaters (Figure 5) which prevent explosive vaporization of water and reduce deaeriation requirements; however, most research has been directed toward a more promising process, the indirect vapor process or closed rankine cycle.

TABLE 1[12]

U. MASS OPEN CYCLE DESIGN SUMMARY (100 $M_w$ output)

| | |
|---|---|
| **CONDENSER** | |
| Tube Diameter | 1 inch nominal |
| Shell Diameter | 46.0 feet |
| Number of Tubes | 137,500 |
| Core Length | 39.1 feet |
| Inside Velocity | 10.0 ft/sec |
| Overall Heat Transfer Coefficient | 692.89 BTU/hr-ft$^2$-°F |
| Total Heat Transfer Area | 1,954,052.94 ft$^2$ |
| Core Pressure Drop | 7.08 psi |
| Cooling Water Pump Work | 12.55 $M_w$ |
| **EVAPORATOR** | |
| Inner Diameter | 50 feet |
| Outer Diameter | 250 feet |
| Film Thickness | 0.010 inches |
| Tiers | 2 |
| Height | 15.52 feet |
| Reynolds Number | 293.3 |
| Heat Transfer Coefficient | 505.6 BTU/hr-ft$^2$-°F |
| Required Area | 41,299,877.6 ft$^2$ |
| Height of Sheet | 5.76 feet |
| Distance Between Sheets | 2 inches |
| Required Pump Power | 27.61 $M_w$ |
| **TURBINE** | |
| Turbine Efficiency | 93% |
| Specific Diameter | 1.525 |
| Specific Speed | 70 |
| Mass Flow Rate | 1.523 x $10^5$ lb/man |
| Outlet Quality | 97.31% |

Table 1 (cont)

| | |
|---|---|
| TURBINE (cont) | |
| Volumetric Flow Rate | $4.2109 \times 10^6$ $ft^3$/sec |
| Turbine Diameter | 235.42 feet |
| Turbine Speed | 80.12 RPM |
| Root Diameter | 151.88 feet |
| Turbine Blade Height | 83.54 feet |
| COLD WATER PUMP | |
| Type | propeller |
| Heat | 17.45 feet |
| Flow Rate | $5.78 \times 10^6$ GPM |
| Outer Rotor Diameter | 38.75 feet |
| Inner Rotor Diameter | 19.38 feet |
| Shape Number | 500 |
| Rotary Speed | 37.6 RPM |
| PARASITIC LOSS | |
| Evaporative Exit Pump | 27.61 $M_w$ |
| Cooling Water Circulating Pump | 12.55 $M_w$ |
| Deaerator Parasitic Work | .61 $M_w$ |
| Fresh Water Pump | .12 $M_w$ |
| Total | 40.89 $M_w$ |
| Net Output (100 $M_w$ Gross) | 59.11 $M_w$ |

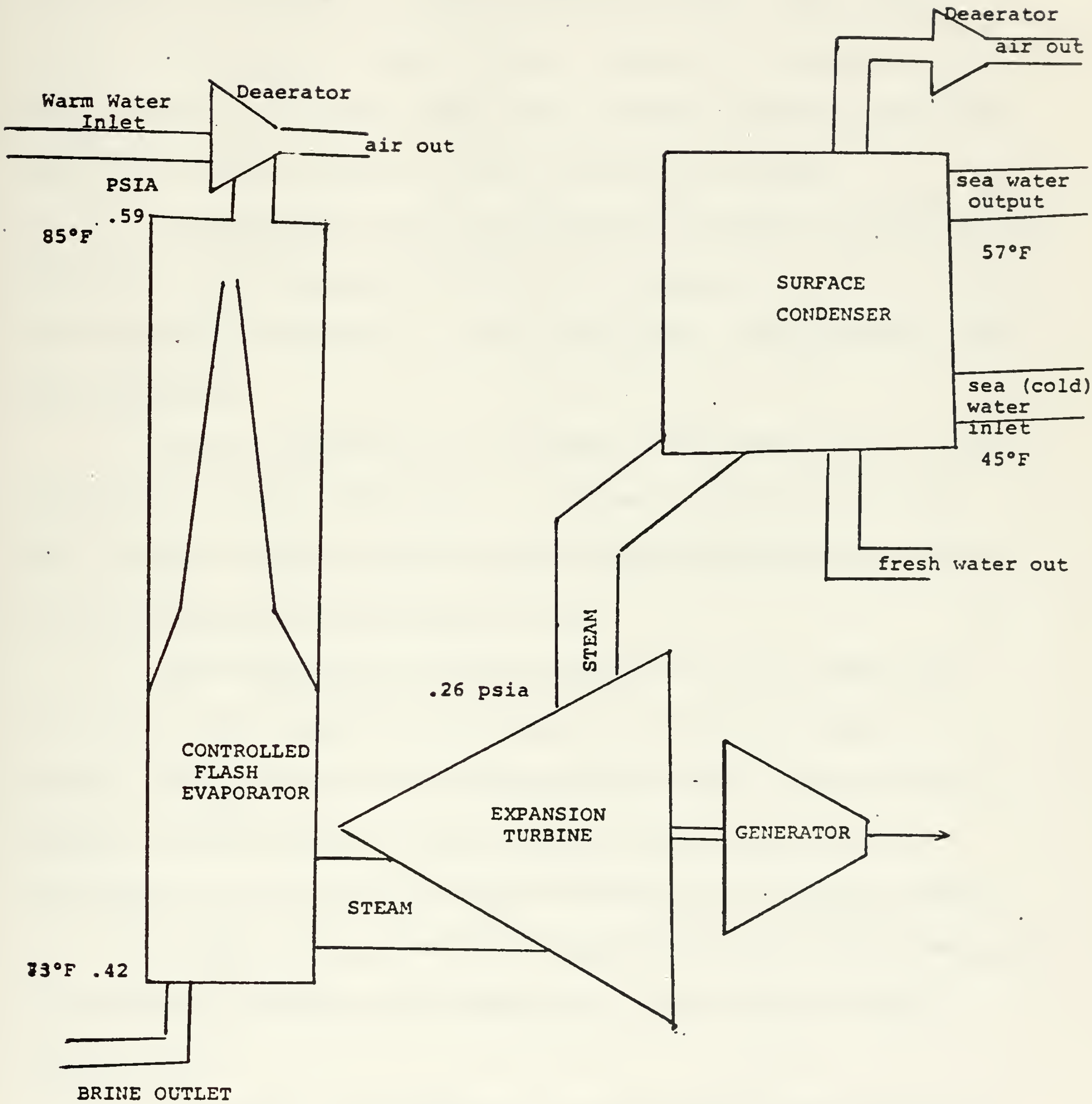



FIGURE 5

CONTROLLED FLASH EVAPORATION SYSTEM (8)

Citing what he believed to be major technological achievements since the Claude experiment, J. Hilbert Anderson in 1966 proposed a far different concept than had been previously attempted. This concept was the ocean cited closed cycle thermal power plant. Anderson concluded that this closed cycle plant was a more viable alternative to the inefficient open-cycle for the following reasons:

a. Sufficient experience had been gained in refrigeration and other closed cycle technologies rendering working fluids, he suggested propane, to be more available, inexpensive, and non-corrosive.

b. Enough theoretical work had been completed to conclude that highly efficient ($\eta_T$ = 90% in Balje Diagrams) gas turbines using these working fluids could be fabricated without excessive development cost.

c. The equalized pressure design of heat exchangers using the outer hydrostatic pressure of the sea water to counter the force of the working fluids vapor pressure would enable the designers to significantly decrease the wall thickness of the heat exchangers saving high material costs.

d. Stable ocean platforms were now being widely employed in increasing numbers and complexity and with very high reliability.

e. Offshore and undersea construction technology and experience had dramatically increased due to oil exploration.

f. Ocean drilling technology could be readily applied to the development of a cold water pipe design.

g. The ability to ocean site the plant would give closer proximity (.4 Km) to cold water of suitable temperature than was available at a shoreline site (2-3 Km). This would result in a decrease of temperature increases which occurred in the longer pipe.

h. Undersea power transmission capability was developed enough so that if a plant could be located close enough to a shoreline, it would be able to readily transfer electrical energy to the local power grid.

i. Modular design of the power units with 25 Mw output reducing turbine sizing and increasing efficiency was feasible.

Although many extrapolations of technology which Anderson assumed have yet to be realized, the effects of his work upon the direction of future OTEC research have been profound.

The development of an economic ocean sited closed cycle system has become the major focus of concern.

Figure 6 shows the schematic of a typical closed cycle system developed by Carnegy-Mellon University. It assumed ammonia was a working fluid. The major components of the cycle being the condenser, turbine generator, and evaporator. The major auxiliary machinery being the warm water pumps, cold water pumps, working fluid feed or booster pump. In the closed cycle system, the working fluid is pumped into an evaporator where it vaporizes at an ambient temperature of about 68°F under high pressure. This high pressure ammonia is forced through a gas turbine where it expands causing this

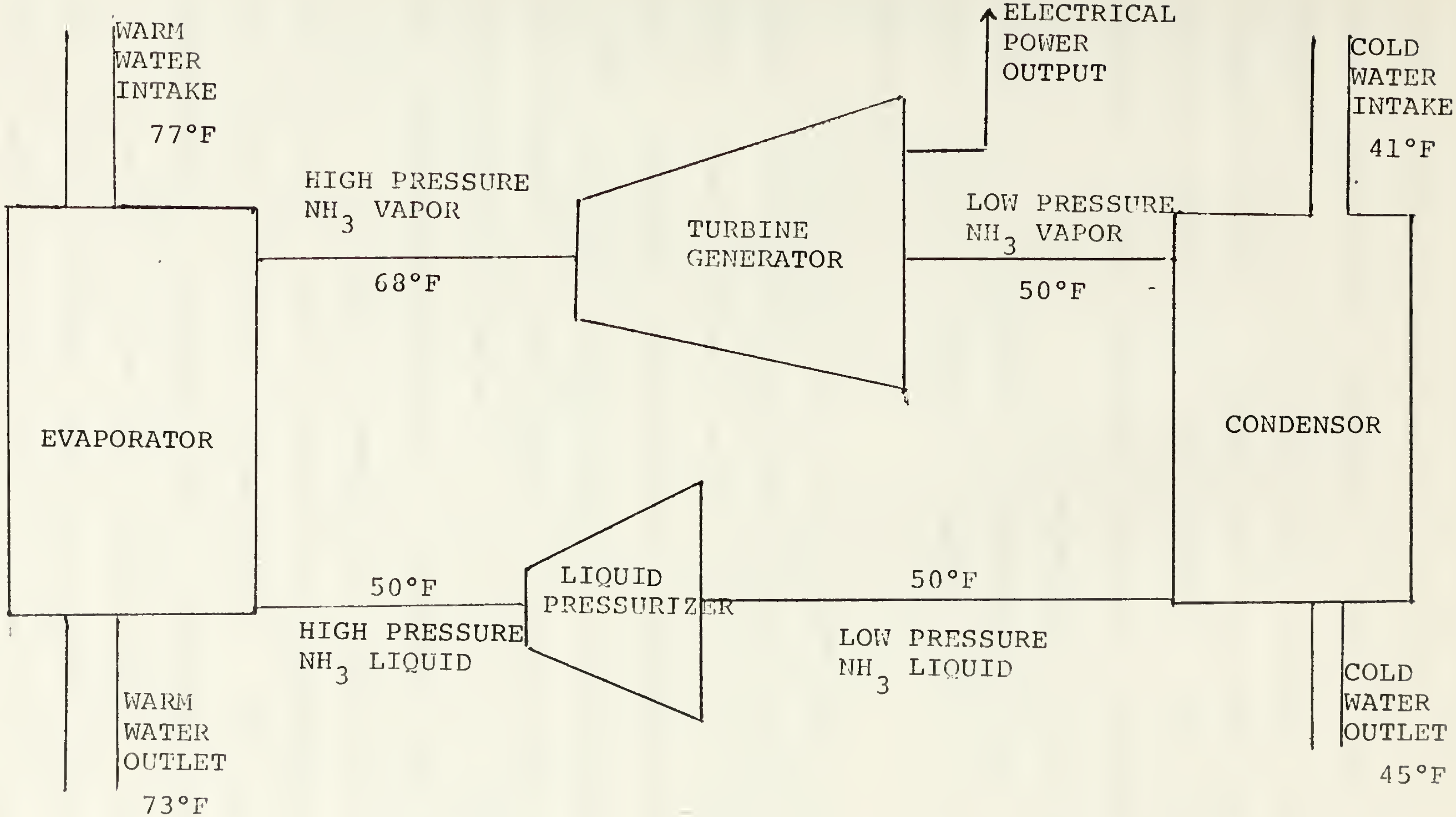



FIGURE 6

SCHEMATIC OF AMMONIA CLOSED CYCLE SYSTEM [17]

termal energy to be transformed into mechanical energy, torque, driving a generator which produces electricity. Once through the turbine, the ammonia now at a lower pressure and temperature, its extractable energy expended, flows through a condenser where it contacts with the tube walls, cooled by the deep ocean sea water and condenses. The condensate is then repressurized and pumped back into the evaporator completing the cycle. Figure 7 is the temperature-entropy diagram of a closed cycle. The absence of superheated vapor should be noted, therefore, although a rankine cycle, it approaches a Carnot cycle.

The closed cycle receives its name from the fact that the working fluid at no time comes in direct contact with the source of heat, the warm water, or the heat sink, the cold water. The heat is transferred by convection across the heat transfer materials of the evaporator and condenser, i.e., the heat exchangers. From Figure 8, it can be seen that total energy available to generate electricity is limited by the total temperature difference of the ocean area where the plant is cited. The change in temperature across the turbine, doing the work, can be maximized by decreasing the temperature losses across the heat exchangers. This can only be accomplished by increasing the heat transfer coefficient on the surface area of the heat transfer units. Consequently, since the heat transfer coefficient remains fairly constant and the mass flow rates of the sea water are extremely high,

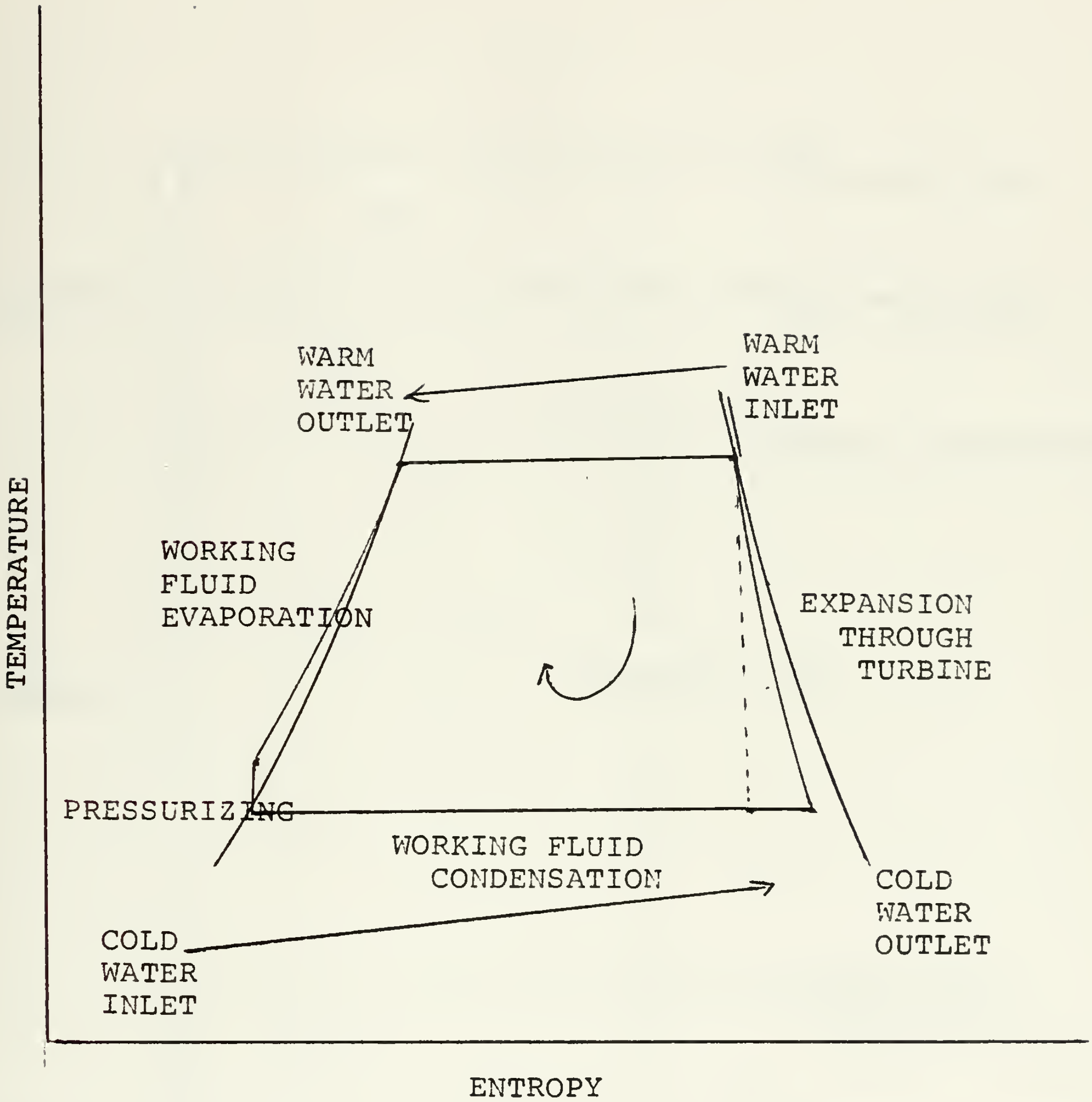



FIGURE 7

OTEC TEMPERATURE ENTROPY DIAGRAM [4]

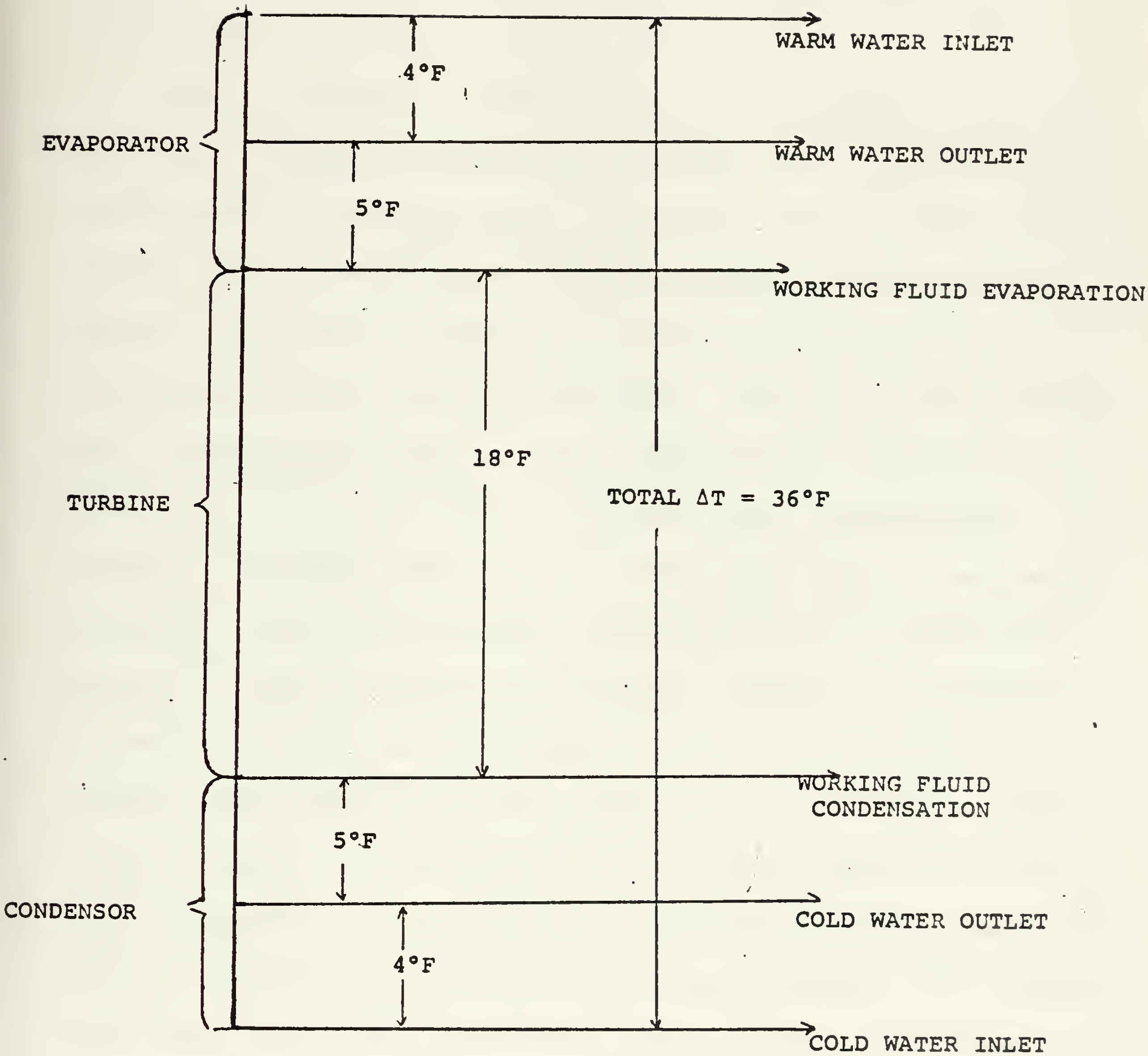



FIGURE 8

TEMPERATURE ALLOCATION [13]

the surface area requirements for a closed cycle system to achieve even this 3% Carnot efficiency are also extremely large. The magnitude of the material requirements for the closed cycle plant heat exchanger is so large that 50% of the capital cost of the entire plant is in heat exchangers. (See Appendix A)

The motivation for the closed cycle selection over the open cycle system in research appeared to be a trade-off for the benefit of reduced parasitic losses due to lower pumping and deaeriation requirements and decreased turbine sizing against the costs of large quantities of working fluids and increased sizing of heat exchangers. Whether these benefits have exceeded the costs of this trade-off is difficult to determine, and is a matter of debate among researchers. If the heat exchanger area of the closed cycle can be reduced by increasing the heat transfer coefficient or by developing methods of heat transfer enhancement without significantly increasing fabrication or natural costs, as is assumed by closed cycle proponents, this trade-off will be of benefit. However, should improvements, such as the controlled flash unit evaporator, succeed in the eliminating of the parasitic losses involved in rendering the vapor suitable for injection into the turbine and should the cost of fabricating and housing a large turbine decrease, the decreased heat transfer surface areas requirements of the open cycle will make it more competitive with the closed cycle. Present technology

appears to favor the development of the closed cycle plant for the same reasons which Anderson assumed. Therefore, design considerations based upon the closed cycle system will be discussed throughout the remainder of the chapter.

## DESIGN CONSIDERATIONS

Scope. The closed cycle system to be developed for ocean thermal energy conversion plans represents a design of an apparatus of a magnitude larger than in existence in conventional or nuclear power plants technologies. Even with the present modest objective of a 100 Megawatt plant, the size of cold water pumping requirements being in excess of 8 million gallons per minute necessitates development of pumps which are larger than any in existance today save some used in the Netherlands to drain large coastal areas. While as in the case of the turbine design, sufficient theoretical work has been completed to suggest that design and development of a turbine suitable for plant requirements can be accomplished within tolerances and required efficiencies, a great deal of work is still necessary to develop an economic heat exchanger, the plants most expensive and expansive component. All design requirements represent extrapolation of present technology not yet applied at this magnitude.

To understand the nature of the OTEC design problem, it is necessary to subdivide the system into its major components: the turbine, heat exchangers, working fluids, auxiliary pumping apparatuses, and hull structures.

Turbine. As stated previously, the component of the OTEC that appears to be the one with least risk involved in its development is the turbine. The turbine design will be influenced by the available $\Delta T$ and will influence other major components notably in system sizing. Figure 9 shows the results of a study conducted by the University of Massachusetts in the optimal sizing of a turbine in regards to its output in megawatts and capital cost in dollars per kilowatt. While there is some debate as to the accuracy of this cost, the economics of the 25 Mw turbine sizing are not. Given these two prespective working fluids, the range of optimal efficiency ranges from 20 to 40 Mw gross turbine output. Beyond these ranges costs increase without yielding adequate returns in output. The 25 Mw design from manufacturers' standpoint is also advantageous since its size and strength requirements do not go beyond what has been shown to be technically feasible. The 25 Mwe net turbine output also lends itself easily to modular construction enabling OTEC to be designed as a combination of independent turbine units, four for a 100 Mw plant, increasing reliability.

In designing a turbine the two most important considerations given that a high efficiency is desired are specific diameter and specific speed of turbine. Specific diameter is defined as

$$D_s = \frac{D \times H^{1/4}}{(Q/60)^{1/2}}$$

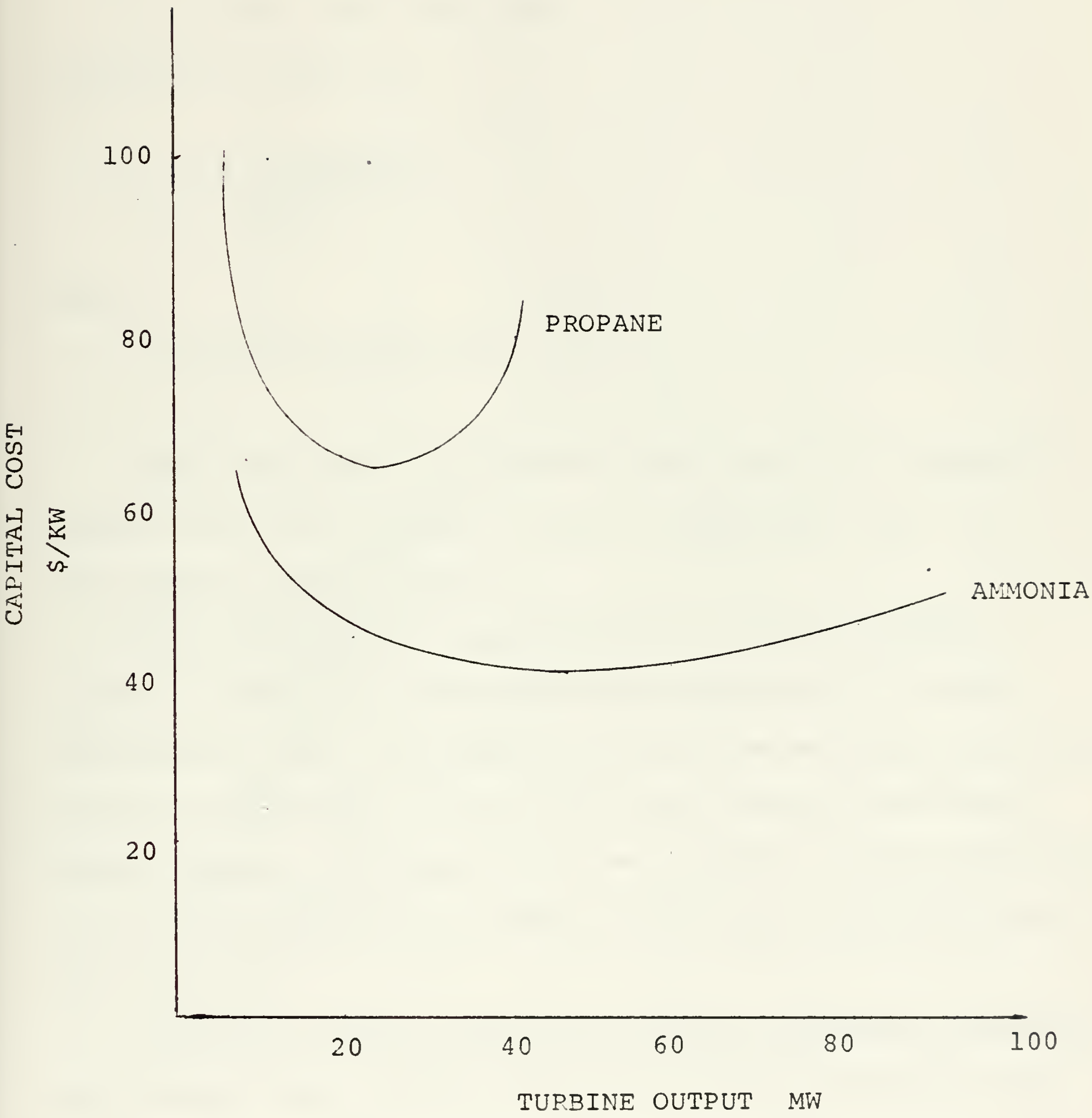



FIGURE 9

RESULTS OF U.MASS (AMHERST) TURBINE SIZING OPTIMIZATION [9]

where,

$D$ = wheel diameter, feet

$H$ = adiabatic (isentropic) head, ft-lb/lb

$Q$ = volume flow rate, CFM

and specific speed, $N_s$;

$$N_s = \frac{N \times (Q/60)^{1/2}}{(H)^{3/4}}$$

where,

$N$ = turbine speed, RPM

These equations are converted from mechanical energy to thermal units and solved in terms of isentropic work (BTU) per pound of working fluid.

Using these design parameters, one can use a Balje diagram, Figure 10, to determine the theoretical turbine efficiency. From this figure it is demonstrated that efficiency of 90% is achievable. For a given efficiency, the optimal turbine design lies along the lower position of the isoefficiency curve with values ranging from $N_s = 80$, $D_s = 1.4$ to $N_s = 100$, $D_s = 1.3$.

Since the specific speed and specific diameter parameters are nebulus terms, another method of turbine sizing has been developed using dimensionless coefficients related to diameter, rotation speed and tip velocity. These were suggested by J. Hilbert Anderson in his work of OTEC design

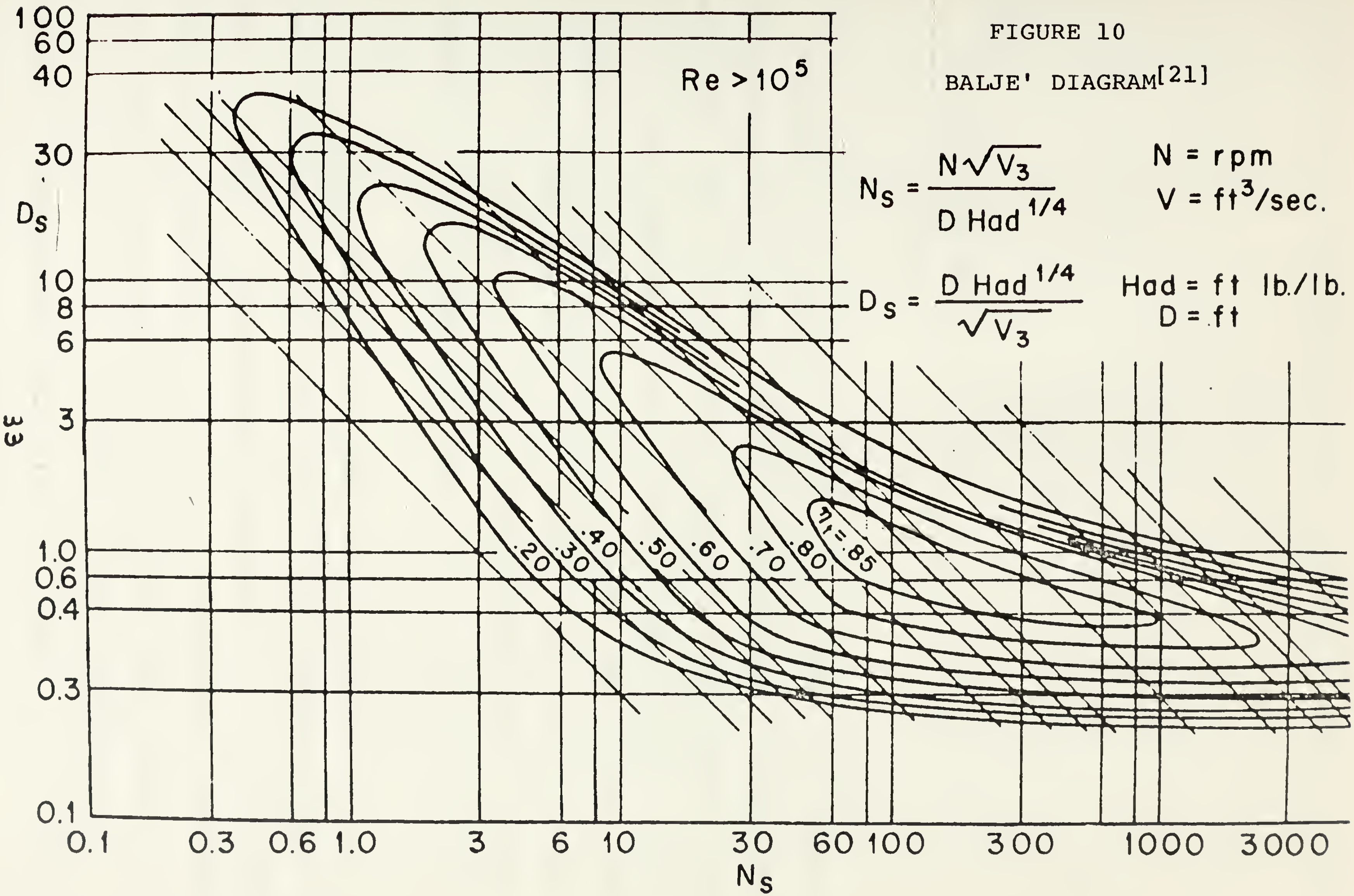



FIGURE 10

BALJE' DIAGRAM[21]

and are the flow coefficient K and head coefficient $K_i$ where

$$K = Q/ND^3$$

$$K_i = 2q\ H/V_{tip}^2$$

where

$q = 32.0$ ft/sec$^2$

$V_{tip} = \pi ND/60$

The flow or capacity coefficient shows how much flow can be achieved for a given diameter turbine while the head coefficient show the head which a given speed and diameter take. For a 25 Mwe turbine driving a generator at 1800 RPM, using ammonia as a working fluid, $K_i = 4.0$, $K = .995$, $D = 62.9$ ins. $N = 1800$, from Figure 11, $D_s = .95$, $N_s = 95$, and from Balje diagram, Figure 10, $\eta_T = 88\%$. The efficiency can be raised to 90% if N is decreased 1661 RPM as $N_s = 80$, $D_s = 1.36$ and $D = 8.0$ ft.

It is important to note that the size of the turbine is not large and the rotational speed is not excessive--equal to that of a steam turbine--while the turbine is very efficient. Also, when other hydrocarbons such as propane or flourocarbon refrigerants are substituted for ammonia, their lower is isentropic work per lb. of working fluids yields larger specific diameters and lower specific speed, i.e., larger turbine operating at lower speeds.

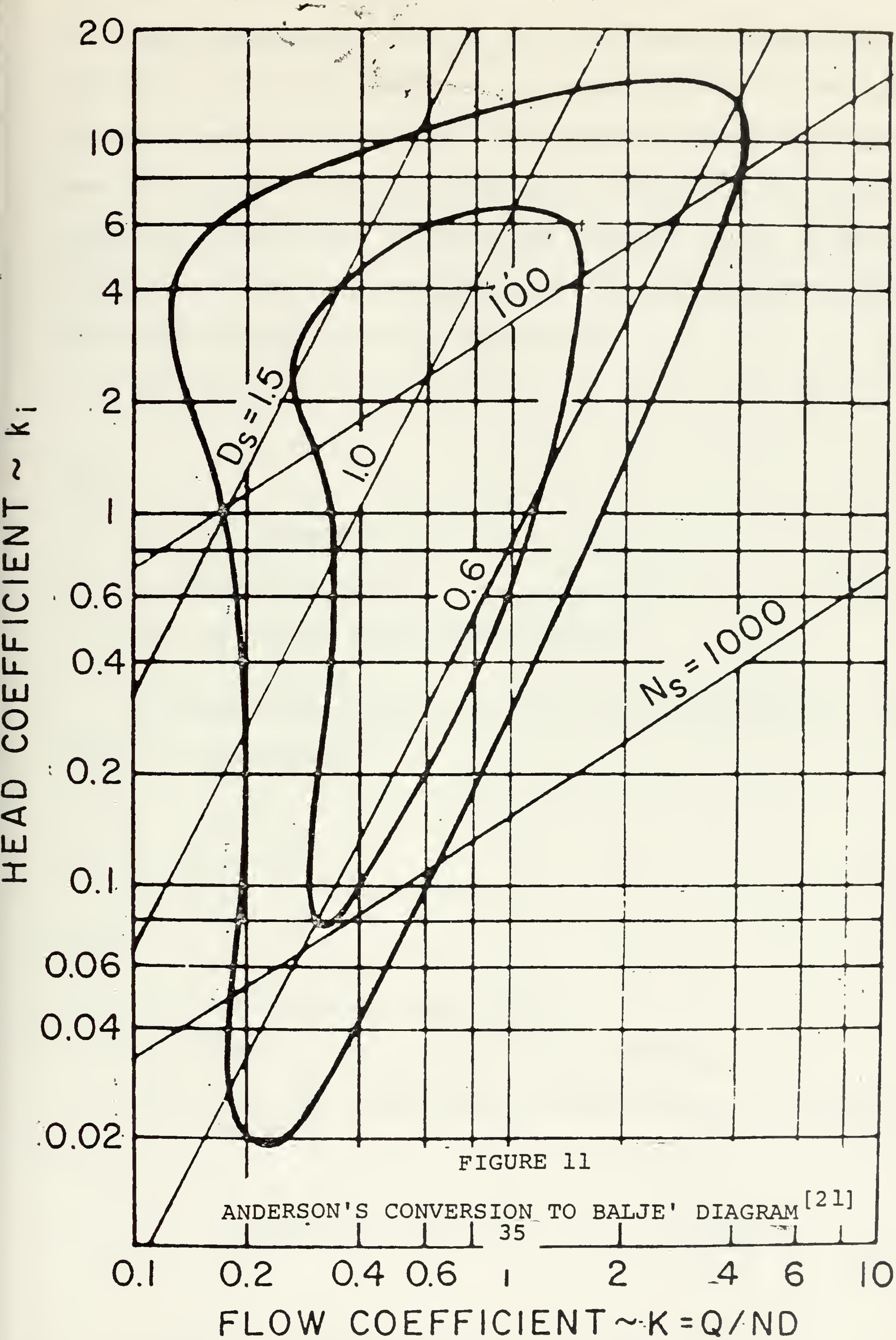



FIGURE 11

ANDERSON'S CONVERSION TO BALJE' DIAGRAM [21]

Heat Exchangers. Due to the low temperature differences available and low enthalphy of the ocean water, a tremendous amount of water both hot and cold must be pumped through the heat exchangers to insure that the working fluid receives as much heat as possible to drive the turbine. For 100 Mw net plant output the required amount of heat is extremely large. This relationship has been defined as

$$Q = \dot{m}\Delta h$$

where

$Q$ = heat transfer rate

$\dot{m}$ = mass flow rate

$\Delta h$ = change of enthalphy of fluids

To express this definition in terms necessary for design, it can be rewritten as

$$Q = U\ A_T\ \ t_m \qquad (1)$$

where

$Q$ = heat transfer rate, BTU/Hr

$A_T$ = total surface area of heat exchanger, $Ft^2$

$\Delta t_m$ = log mean temperature across the heat transfer surface

(refer to Figure 7 $\quad \Delta t_m = \left(\dfrac{\Delta t_H - \Delta t_C}{\ln \dfrac{\Delta t_H}{\Delta t_C}}\right)$

$U$ = heat transfer coefficient BTU/Hr-$Ft^2 \cdot F°$

For design, area is of interest, equation (1) becomes:

$$A_T = Q/U\ \Delta t_m$$

Using available experimental data and solving, $A_T$ (total heat exchanger surface area), has been found to be in the range of 10-40 $ft^2$/kw or as high as $4.0 \times 10^7$ $ft^2$ of heat exchanger material. Therefore, due to the high flow rates and low temperature difference, the surface area of the heat exchangers is large and costly, representing the largest single item expenditure for an OTEC plant. At this embryonic stage of OTEC development, they are the pivotal obsticles to more extensive considerations.

Research in OTEC heat exchanger design has cited, among other considerations, the following of primary importance: sizing, geometry, heat transfer coefficient, location, interaction with working fluids, effects core pressure drop within the heat exchanger, and the effects of environmental agents.

The sizing and geometric considerations are important in the constraints which they place upon the overall OTEC design such as the hull structure, and in the means which they vary other design considerations such as the heat transfer coefficient. Various experiments have been conducted using both the traditional tube-shell arrangement (Figure 12) and the plate-fin types (Figure 13) to increase the overall U for the heat exchangers. The plate-fin geometry has been found to have a superior heat transfer coefficient. Size and geometry

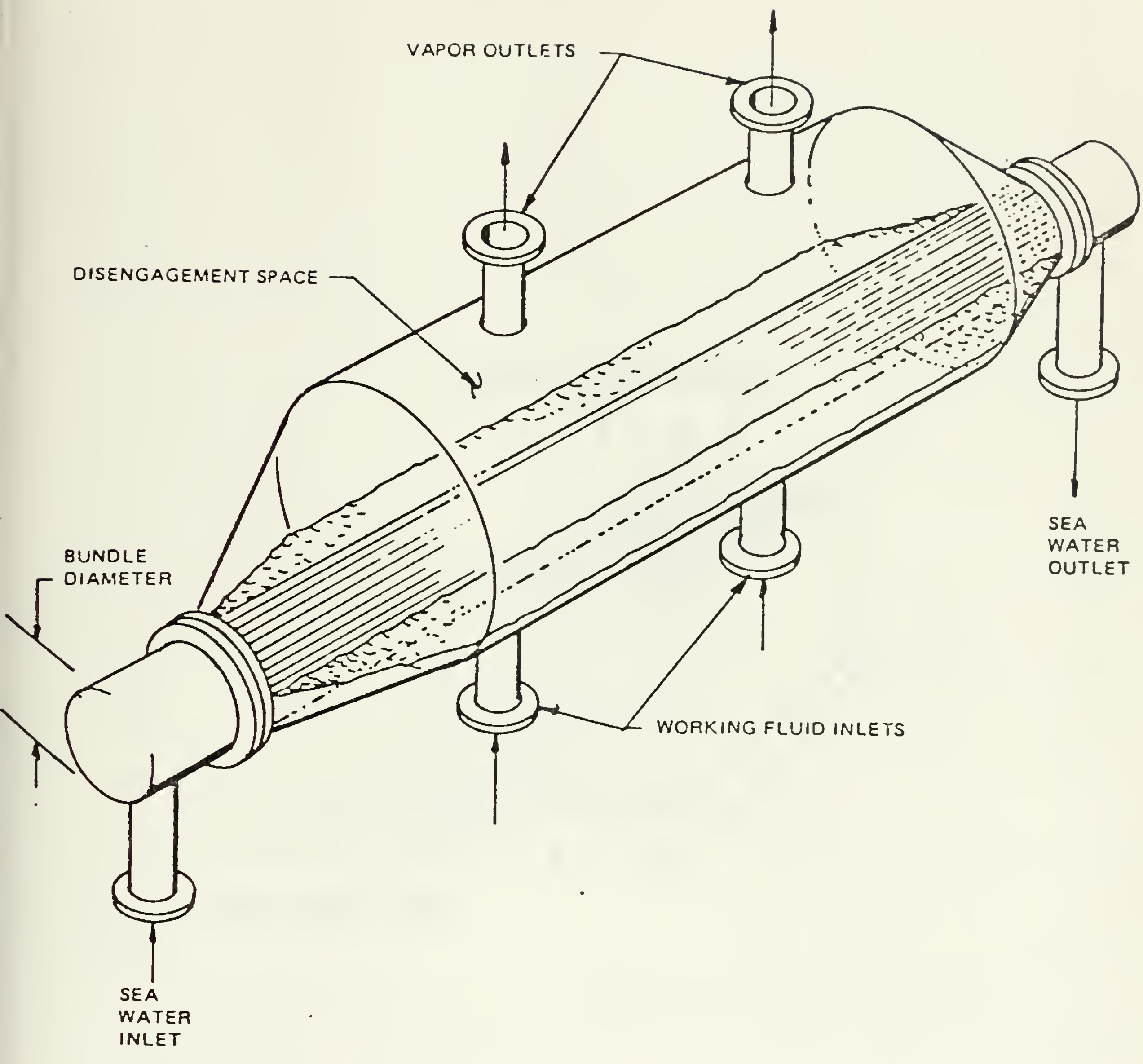



FIGURE 12

TUBE-SHELL TYPE HEAT EXCHANGER [21]

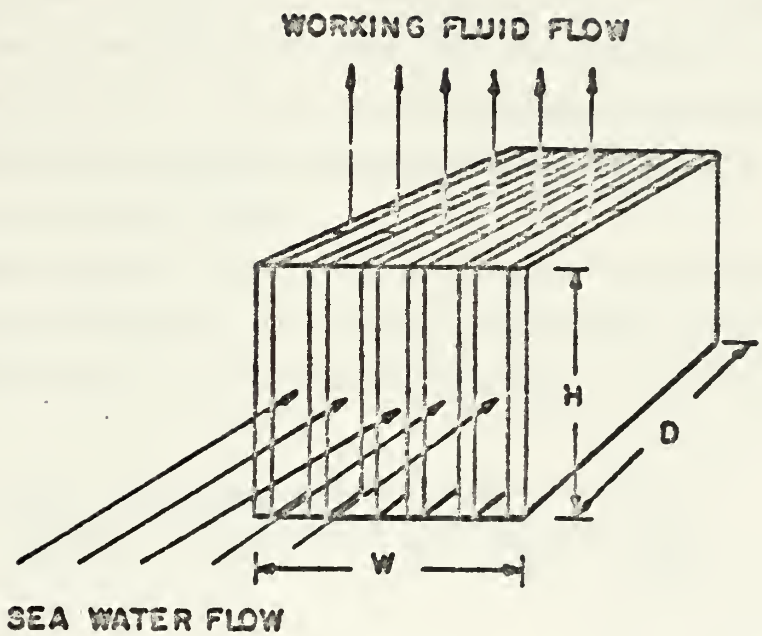



FIGURE 13

PLATE-FIN HEAT EXCHANGER CONFIGURATION [9]

will also affect the fabrication cost of heat exchangers with the more complicated arrangements, possibly adding to costs in excess of the benefits in decreased surface area which they confer. Certain geometric arrangements can also increase surface area without a significant increase in material requirements as the arrangement of fins within the tube demonstrates. (Figure 14). Finally, sizing and geometry are important in providing not only the most efficient use of surface area but in giving the heat exchanger strength and structural rigidity necessary for the hydrostatic loads which will encumber them.

From equation (1), it can be seen that U, the heat transfer coefficient, is inversely proportional to the required surface area of the heat exchanger. U is defined as

$$U = \frac{1}{\frac{D_o}{D_i h_i} + \frac{D_o \ln(D_o/D_i)}{2K} + \frac{1}{h_o} + \frac{D_o}{D_i h_{d,i}} + \frac{1}{h_{d,o}}} \qquad (2)$$

where

$U$ = overall heat transfer coefficient of heat exchanger

$D_o$ = tube outside diameter

$D_i$ = tube inside diameter

$K$ = exchanger thermal conductivity

$h_o$ = outside tube surface-heat transfer coefficient

$h_i$ = inside tube surface thermal heat transfer coefficient

$h_{d,i}$ = inside tube surface heat coefficient subject to fouling

$h_{d,o}$ = outside tube surface heat transfer coefficient subject to fouling

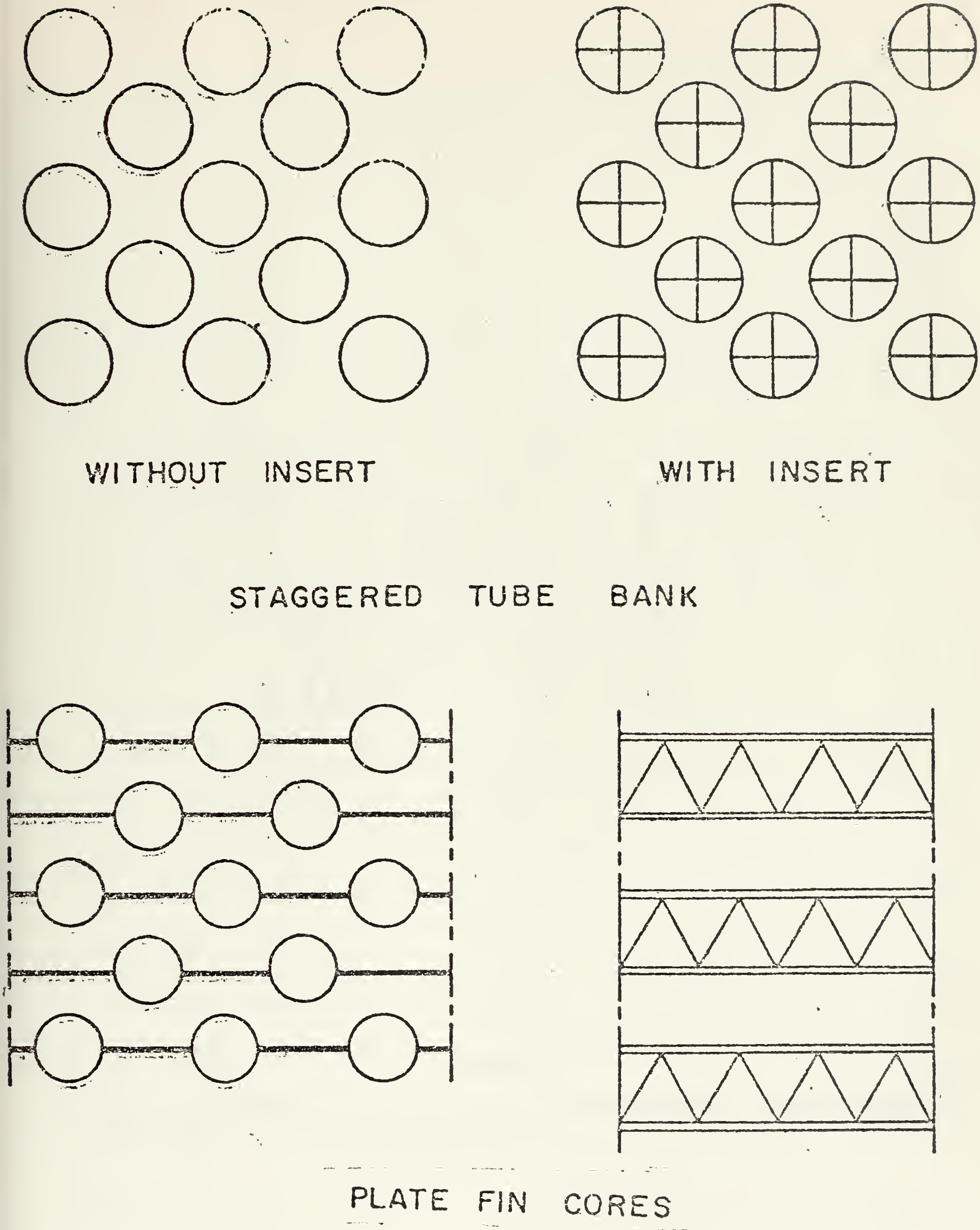



FIGURE 14

METHODS OF SURFACE AREA ENHANCEMENT [21]

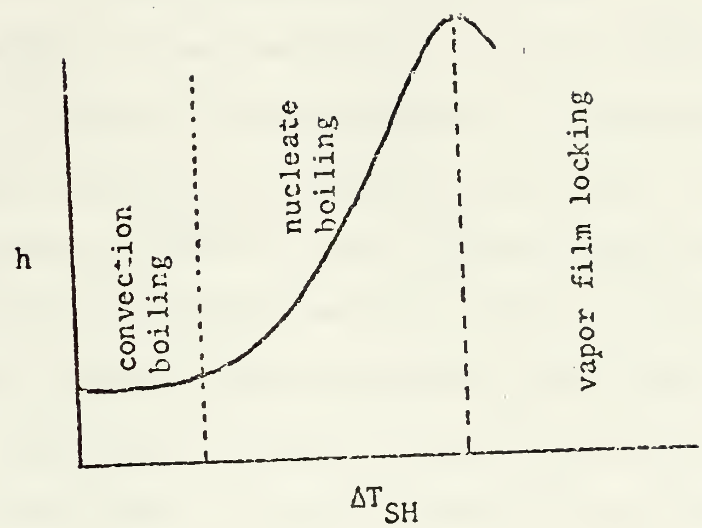



FIGURE 15

EFFECTS OF TYPE OF BOILING UPON HEAT TRANSFER COEFFICIENT [17]

By increasing U, the surface area $A_T$ can be reduced. Due to the magnitude of $A_T$, this reduction can have a profound impact upon the overall cost of the heat exchanger and, as has been theorized, an even greater impact upon overall shore plant cost. The relationship of the heat exchanger cost to overall cost is such that a decrease in heat exchanger cost results in a reduction in overall plant cost greater than the decrease in heat exchanger cost alone.

One method of increasing the overall heat transfer coefficient is through the enhancement of the tube surface thermal heat transfer coefficient $h_i$ or $h_o$. In the evaporator this has taken the form of tube surface preparation which forces evaporation of the working fluid to occur in nucleate boiling with a higher heat transfer coefficient rather than in normal convection of pool boiling with a very low heat transfer coefficient as usually occurs in an evaporator. Figure 15 demonstrates this situation. In condensation, $h_o$ is fairly high to start with but will deteriorate rapidly with accumulation of working fluid condensate upon tube walls. To keep $h_o$ for condensation high plus to utilize all available surface area in a condenser, corrigation of the tube surface with the condenser tube stacked vertically has been suggested by Dr. Lavi of Carnegie-Mellon University. Figure 16 demonstrates this arrangement. Notice also that condensate flows through the tube in a wave action pulled vertically by gravity. This has the effect of wiping the condensate accumulation from the tube wall maintaining a film in the walls

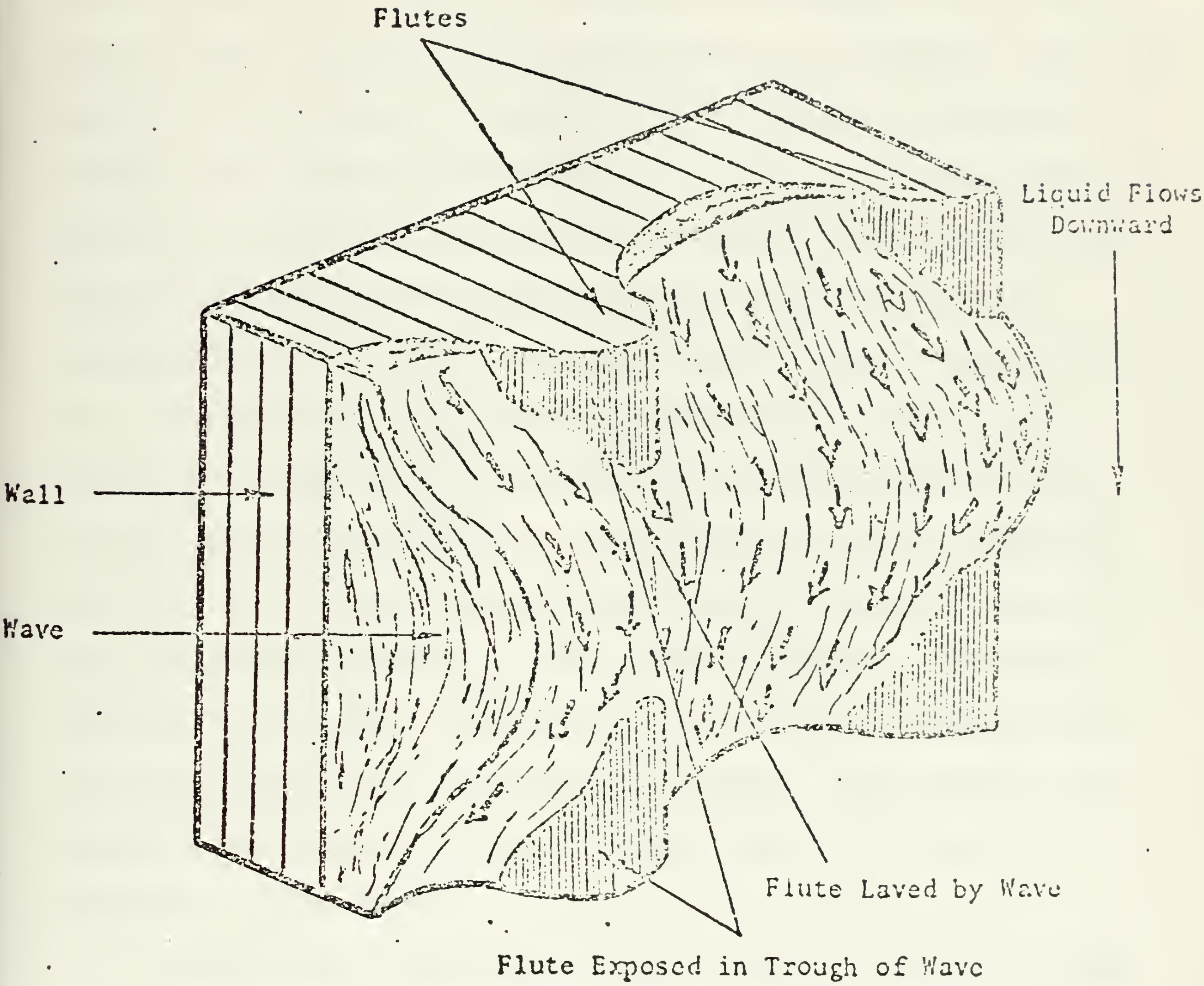



FIGURE 16

FAILING FILM WAVE MOTION ON GUTTERED SURFACE CONDENSER[21]

throughout the condenser, a most efficient means of heat transfer.

The locations of the heat exchanger are important for several reasons. One of the primary considerations is the effect which location of the evaporator or condenser has upon pumping requirements. Because of the separation existing between the warm and cold water, it was once surmised that location of the particular heat exchanger units within the ocean zone of the sea water with the required injection temperature would reduce pumping requirements. Analysis of this arrangement later demonstrated that the additional parasitic losses due to the handling of the working fluid between the heat exchangers were greater than the benefits received by reduced pumping requirements. Another impact of heat exchanger location is upon tube thickness. Refering to Table 2, we see that internal vapor pressure of the working fluid can neutralize hydrostatic loading. The smaller the pressure difference between the tube sides, the lower the thickness requirement.

How the heat exchangers reacts with the working fluids will determine the type of materials to be used. In equation (2), it can be seen that material and working fluid selection affected every factor which determines overall heat transfer. Natural strength and corrosion suseptability will determine $D_o$ and $D_i$. K is a direct function of the material itself, $h_o$ is a function of the working fluid and $h_{d,o}$ is a function of both. $h_{d_i}$ is a function of the material and the ocean environment.

TABLE 2

WORKING FLUID PROPERTIES[19,17]

| WORKING FLUID | IDEAL CYCLE EFFICIENCY % | CYCLE EFFICIENCY 5% ΔP/P | PRESSURE PSIA 70°F | PRESSURE PSIA 50°F | DEPTH FOR BALANCING VAPOR PRESSURE CONDENSER | DEPTH FOR BALANCING VAPOR PRESSURE EVAPORATION | CONDUCTIVITY BTU/ft-hr-°F | IDEAL MASS FLOW RATE lb/min |
|---|---|---|---|---|---|---|---|---|
| Ammonia | 3.72 | 2.71 | 128.8 | 89.2 | 170 | 260 | .29 | 317,600 |
| R-12 | 3.68 | 2.57 | 85. | 61.4 | 107 | 160 | .04 | 2,630,000 |
| Propane | 3.72 | 2.82 | 124.7 | 92.2 | 179 | 252 | .056 | 1,084,000 |

The object of the design would be to use the material with the highest thermal conductivity as modified by strength and cost with a working fluid of highest heat transfer properties if compatibility were not a serious problem which it is. Table 3 demonstrates the superior performance qf 90 Cu 10 Ni alloy in reducing material requirements yet this material is most suseptable to the corrosive action of the most efficient working fluid, ammonia. The service life of such a heat exchange would be uneconomically short. The same problem exists with 5083-0 Aluminum which is more competitive than 90 Cu 10Ni due to its reduced cost. If one decides to use these structural materials, another working fluid such as propane must be used with all the disadvantages which it possesses. If, on the other hand, ammonia is selected as a working fluid, more expensive materials such as titanium must be used. The benefits of such trade-off are difficult to determine because of the extensive implications of the trade-off in all areas of the OTEC design.

Figure 17 presents the effect upon overall cycle efficiency due to heat exchanger pressure drop. Notice, that at 5% this becomes very dramatic. This is determinant of heat exchange geometry and therefore a constraint in design. Core pressure drops are also important as they represent a loss in the condenser which must be overcome by the cold water pump. These losses are defined as follows:

TABLE 3

EFFECTS OF MATERIAL SELECTION UPON HEAT EXCHANGER SIZING[9]

| MATERIAL | 26.7 Mw BOILER MODULE [m] | 25 Mw CONDENSER MODULE [m] | TOTAL VOLUME REQUIREMENT [$m^3$] |
|---|---|---|---|
| Aluminum (5083-0) | 9 x 23 x 11 | 10 x 18 x 11 | 3511 |
| Copper-Nickel (90/10) | 9 x 22 x 11 | 10 x 19 x 11 | 2865 |
| Titanium (TI-75A) or Inconel 625 | 9 x 27 x 11 | 11 x 22 x 11 | 3473 |
| Plastic | 29 x 25 x 22 | 16 x 16 x 22 | 24,770 |

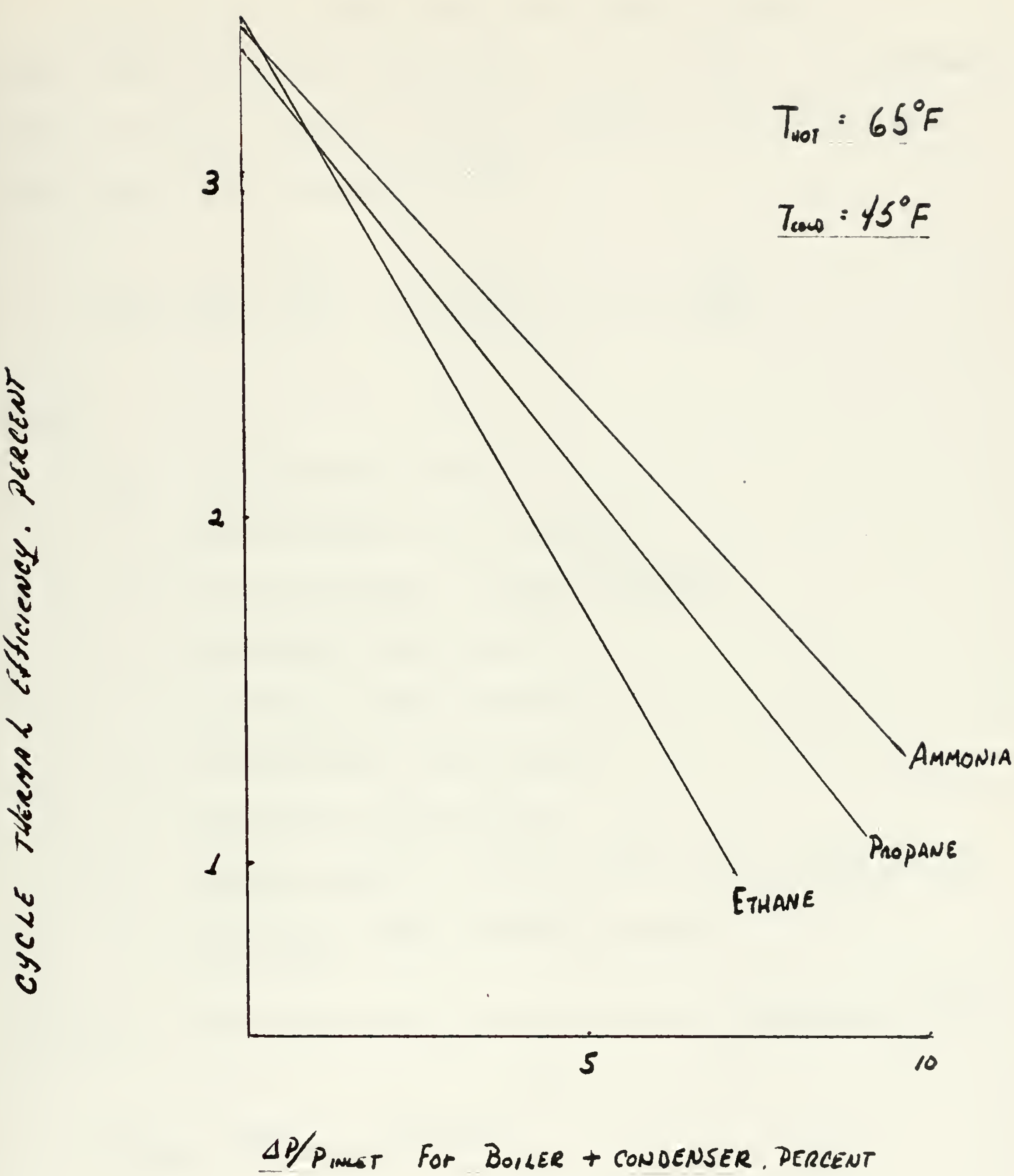



FIGURE 17

CYCLE EFFICIENCY VS. PRESSURE DROP [4]

total loss = inlet tube loss + core losses

Inlet tube losses are a function of cold water tube geometry, water velocity, gravity, and the tubes loss coefficient. Core losses are defined as:[12]

$$\text{core loss} = \frac{\Delta P_c}{\rho} = (f\ L/D + K_c + K_e)\ \frac{V^2}{2g_c}$$

where

$\Delta P_c$ = core pressure drop

$\rho$ = water density (64.0 $lb/ft^3$)

$f$ = friction factor (Moody diagram)

$L$ = condenser tube length

$D$ = condenser tube diameter

$K_c$ = contraction coefficient

$K_e$ = expansion coefficient

$g_c$ = gravitational constant = 32.2 $lb_n\text{-}ft/lb_f\text{-}sec^2$

$V$ = condenser tube inside velocity

Since the cold water pump requirements to overcome losses is

$$\dot{W}_{pump} = (\dot{m})_{cold}\ (\text{losses})$$

where

$\dot{W}_{pump}$ = pump power to overcome losses

$\dot{m}$ = mass flow rate of cold water

losses = total loss

Reducing the core loss will reduce the overall pumping required for the cold water pipe and thereby reduce parasitic losses of the plant. This will entail variations in heat exchange geometry constrained by the minimum surface area parameters.

The heat exchangers will operate in an ocean environment which is full of both corrosive and fouling agents. Biofouling and, to a lesser extent, corrosion represent unknowns which have a potentially detrimental effect upon the heat exchanger's service life and heat transfer coefficient. An extensive pool of knowledge is available to assist in the selection of methods to keep corrosion within tolerance, and material selection will determine the degree to which corrosion becomes a problem. It is known that certain materials such as 90 Cu 10 Ni are toxic to marine life and therefore will not foul. Other materials such as aluminum and titanium are subject to the possible effects of biofouling. As pointed out in equation (2), a reduction in the overall heat transfer coefficient U can result if a tube surface becomes coated. This will reduce the heat transfered across the heat exchanger and thereby reduce plant output. Given the already low temperature difference, such an occurrance could make an OTEC plant uneconomical to operate and maintain.

Several methods have been proposed to check biofouling besides the one previously discussed. Some methods use injection of a toxic material, such as chlorine, into the

sea water side of the heat exchanger, however, detrimental factors to the ocean environment and crew safety could exist. Possible high rates of injection might be required increasing plant operating costs. Another method involves mechanical scrubbing of tube sides on a periodic basis. This method appears to be costly and limits plant output during cleaning. It has also been shown experimentally that water flow rates in excess of 5 feet per second will also prohibit biofouling material accumulation. If this is true the problem of biofouling becomes a minor one as design volume flows already exceed this velocity.

Pumping Apparatus. Regardless of the location of the OTEC plant, on land or at sea, either as a floating plant or submarine, the requirement of bringing the warm and cold waters together in the heat engine will necessitate an extensive amount of pumping. While early research attempted to use ocean currents to eliminate warm water pumping requirements for the evaporator, it has not been completely successful. In any event no natural current exists with the exception of upwelling which can be used to bring the colder water to the near surface location of the condenser. The pumping requirements for an OTEC plant are very large.

To give a proper prospective of the magnitude of the pumping flows in gallons per minute, the calculation of Dr. Zener of Carnegie-Mellon University are here utilized:

$$\text{Power produced} = \eta_{Car}\ \Delta T_{er}\ C_v\ \dot{M}^{[17]}$$

where

$\eta_{Car}$ = Carnot efficiency - 3%

$\Delta T_{er}$ = available temperature accross the evaporator - (50% entire T available (10°C))

$C_v$ = specific heat of water at constant volume (1 Cal/gm °C)

$\dot{M}$ = mass flow rate (gm/sec)

Inserting appropriate values and converting from calories to joules

$$\text{Power produced} = [(.03)(5°C)(1\ \text{Cal/gm °C})(\dot{M}\ \text{gm/sec})]$$

$$= .15\ \dot{M}\ \text{Calories/sec}$$

Converting to Joules

$$= .15\ \dot{M}\ \text{Calories/sec} \times 4.2\ \text{Joules/calories}$$

$$= .63\ \dot{M}\ \text{Joules/sec} = .63\ \dot{M}\ \text{watts}$$

for 1 kw gross output and solving for $\dot{M}$

$$\dot{M} = \frac{1000}{.63}\ \text{gm/sec}$$

which Dr. Zener states corresponds to 1.65 liters/sec volumetric flow in the CMU design.

For 130 Mw gross output, the volumetric flow will be,

$$\dot{V} = 1.65 \text{ litre/sec} \times \frac{130 \times 10^6}{10^6}$$

$$= 214.5 \text{ liter/sec}$$

converting to gallons,

$$\dot{V} = 214.5 \times 10^5 \text{ liter/sec} \times .264 \text{ gals/liter}$$

$$= 56628 \text{ gal/sec} \times 60 \text{ sec/min}$$

$$= 3.39 \times 10^6 \text{ GPM}$$

This pumping requirement would be for each of the two heat exchangers. For 100 Mw net output, assuming 30 Mw parasitic loss, OTEC will be pumping at least 6 million gallons of water per minutes. If the pump drives can be 90% efficient, then at 30 Mw power, their required horsepower will be

$$\frac{30 \times 10^6 \text{ watts}}{.9 \times 750 \text{ watts/HP}} = 45{,}000 \text{ horsepower}$$ [16]

Considering that these pumps will have very low head pressure (approximately 15-17 ft. for the cold water pump) with low speeds and wide diameters, the technology necessary to develop the pumps will approximate ship propeller design

rather than conventional pump design unless a modular design technique is used.

The pumps required to boost working fluid pressure after condensation are in existance although some redesign might be necessary to make them more compatable with the working fluid.

Working Fluids. The selection of a proper working fluid for use within the OTEC heat exchangers will be dependent upon the following criteria: temperature range of condensation and evaporation, pressure of evaporation and condensation, heat transfer and thermodynamic properties, speed of sound, compatibility of fluid with turbine and heat exchanger materials, overall plant safety considerations, availability of working fluid in large quantities, cost, and environmental impact in event of a leakage.

A working fluid for OTEC will be expected to evaporate in temperature ranges of 70°F to 80°F and condense within a temperature range of 40°F to 50°F. Within these ranges can be found suitable numerous flourocarbons, hydrocarbons, and ammonia so a list of available candidates is fairly easy to compile for the working fluid.

Since most proposed designs of OTEC plants recommend siting the plant at sea, hydrostatic pressure will exist on the water side of the heat exchangers. In order to minimize the required amount of material used in the tube walls of a heat exchanger, a pressure balance might also be adviseable. While other factors such as corrosion might override this

consideration, every effort which could reduce the heat exchanger size should not be discarded. Unfortunately, the lower hydrostatic pressures exist in the evaporator where the working fluid pressure is highest while the higher hydrostatic pressure is opposed by the lower pressure in the condenser. This happens due to the location of the evaporator and condenser nearest to the warmer and colder sea water injection temperatures respectively reducing temperature losses and pumping requirements. Table 2 illustrates these characteristics.

The heat transfer and thermodynamic properties of a fluid will affect turbine design and overall plant efficiency. The fluid's thermal conductivity will determine the flow rates of warm and cold sea water in the heat exchangers required to reduce temperature losses at the working fluid interface. The greater the flow rate the lower plant net output. The thermodynamic properties, speed of sound and heat of evaporation, will determine turbine tip speed limits and volume flow rates. The speed of sound represents a constraint in the upper limit of turbine rpm as efficiency degradation occurs rapidly if tip speed exceeds the sonic speed. In studies using the major candidates, propane, ammonia, and R12/31 flourocarbon, this constraint has not been a binding one. Sonic speed of an ideal gas is defined:

$$V_s = \sqrt{\gamma\, g_o (1545/M)\, T}$$

where

$\gamma$ = ratio of specific heats

$g_o$ = 32.2 ft/sec$^2$

M = molecular weight, lb/mole

T = °R (absolute)

and turbine tip speed;

$$V_{tip} = \frac{\pi}{60} (N_X D)$$

where

N = rotational speed, RPM

D = wheel diameter, ft/sec

A higher heat of evaporation will lower volume flow rates required for turbine output--the lower the volumetric flow rate, the smaller the designed turbine, consequently, some material cost savings can be realized. Another heat transfer factor, mass flow rate, will determine working fluid pumping capacity and corresponding parasitic losses.

If ammonia is selected as a working fluid, its corrosive nature becomes a critical factor in material selection for heat exchangers. Should dry ammonia become, through a leak in a tube wall, contaminated by water, it will form a highly corrosive agent to any heat exchanger constructed with copper metal alloy (90 Cu 10 Ni) or aluminum. This would result in a

drastic shortening of the heat exchanger's surface life. With the most extensive and successful technology of heat exchangers existing with those of copper-nickle alloy, this might be a serious trade-off for ammonia's benefits. More expensive steel apparatus must be used. Flourocarbon or propane would be more compatible with copper alloy, however with propane turbine internal lubrication problems might develop. This compatibility problem has a significant effect upon plant economics.

With some exceptions, the OTEC plants will be manned vessels and therefore will be subject to safety regulations. Ammonia is a highly toxic, flammable, and corrosive chemical in concentrations. A major leak could cause a complete shut down and abandonment of an OTEC power plant and require expensive measures to reopen it. Considering the high capital cost of such a plant, this is a serious risk. Likewise, propane, while neither as toxic nor corrosive, is highly flammable. The flourocarbons pose none of these problems. As for minor leaks, it would be easier to detect the presence of ammonia because of its strong odor than the other fluids when in light concentrations.

It has been estimated that an OTEC plant will require as high as 7 million gallons of working fluid within the closed system to initiate and maintain operations. With this great a concentration of industrial chemicals in an ocean environment, environmental considerations must be acknowledged and dealt with. In the event of a spill, it is unlikely that ammonia

or propane will have a major detrimental impact, but this is not known with certainty. As for the flourocarbons, one has only to observe the present controversy over aerosols using freon propellants and ozone layer depletion to gauge this potential controversy.

Provided the problems of compatibility and heat transfer properties can be rectified, final economic criteria must be applied, these being cost and availability on the market. The costs of ammonia are only a fraction of that for propane and flourocarbons. For the OTEC program to have a significant impact upon the U.S. energy needs, numerous OTEC plants must be built. This represents a tremendously high amount of working fluids. It is doubtful that taking this much flourocarbons from the market is feasible or desirable.

Hull Structure and Configuration. The encasement of the power plant poses the most complex problem in the development of the OTEC concept. In terms of uncertainty, it entails by far the most risk for a potential investor. Although a great deal of work in ocean sited structures has been pioneered by the oil industry in exploring such areas as the Carribean and North Sea, the magnitude of the OTEC plant poses special problems not previously encountered. When one considers that a major structural component of the plant, the cold water pipe, will extend in excess of 1500 feet below the ocean surface, the problem becomes clear. OTEC plants will be structured

with the deepest draft yet constructed for ocean use, neglecting oil drilling devices with only minor structural appendages protruding to deeper depths.

Placing the size of the OTEC plant aside, there are similarities in design considerations of the hull structure with other ocean sited structures which are germain to OTEC hulls. Siting in the ocean assumes that a structure will, in all probability, not be built in the same location as its intended area of operations. Weather, ocean waves, and other environmental factors will not permit long at-sea construction operations to continue without frequent interruption or even costly damage requiring further rework. Special construction equipment requirements will necessitate near land or on land work to avoid costly delays and barge fees. For a large structure the quantity of the construction materials needed to build it might in itself add considerable lighterage fees significantly increasing the capital costs. Finally, the availability of trained construction personnel might be such that at-sea construction will be unfeasible.

Since the OTEC plant will not be built on the site of operation, any design must also take into account the particular nature of plant deployment. During this phase of development, the plant will be subjected to both static and dynamic loading which will not be encountered during normal operations. Availability of tugs in a particular horsepower range and barge sizing may necessitate modular or sectional construction with

on-site assembly. Coastline topography will limit the depth of the construction, and bouyancy and stability requirements will further constrain the design.

Once on site, the environmental forces during plant operation will become the overriding concern. The OTEC design will then reflect the effects of such forces as normal wave loadings, the exceptional "100 year" wave height, and the other normal and adverse weather factors such as wind loadings. If the plant is to be submerged, hydrostatic and wave influences of deep sea loadings must be considered. Further loading considerations must be the shear forces upon such numbers as the cold water pipe due to currents and the dynamic responses of the plant in the ocean environment.

The material used in construction will be influenced by the conditions at the site. Since OTEC represents a large capital investment, the plant design life must be long, in excess of 40 years. This material must be able to withstand the deleterious effects of seawater corrosion and marine life fouling as well as the impact loading forces exerted by ocean waves. Since a great deal of this material will be used in construction, it must be fairly inexpensive and its use in construction condusive to labor and time saving construction methods. In OTEC, reliability of the material will be of paramount concern.

Finally, on-site conditions will drive the design of OTEC's hull configuration. OTEC has buoyance and stability requirements due to the nature of its task which are not

similar to those of any at sea structures. It has special hull sizing requirements to prevent effluent recirculation causing plant efficiency to decrease or even causing ceasation of plant operation. Since the major parasitic power losses in OTEC come from internal pumping demands, hull configuration must be varied to reduce these losses.

What has been discussed above summarizes the major problems facing an OTEC hull structure designer. For a design to be successful, it must be comprehensive in three areas. The OTEC plant will be designed for its construction, deployment, and operating environments. These environments contain separate and distinct characteristics which drive the design. The OTEC design is an iterative process of these major concerns and therefore its feasibility in totality is contingent upon these phases of design being within the environmental parameters.

Prior to what will be discussed later in the baseline designs, most research work has been geared to a design using at-sea operation factors primarily. While the final design of an OTEC plant will most assuredly not resemble those preliminary designs for a variety of reasons, these designs are presented as they represent the first step in the iteration process.

Considering alternative hull configurations, original research work formulated a structure conceived upon locating the power plant in a major ocean current, the Gulf Stream, or in a stagnant pool in a tropical sea. While both of these approaches did require existance of ocean currents to remove

used effluent from the structure, the different environments resulted in radically different hull configurations. To locate in the Gulf Stream would necessitate that the hull not only be structurally suitable but also hydrodynamically sufficient. In order to reduce parasitic power used the plant would have to be anchored in the Gulf Stream and would be subjected to tremendous drag forces if the hull was not properly configured. Positioning could be an economically tenuous affair. Therefore it was proposed that an OTEC plant be designed as a submersible with exposed evaporators anchored in the Gulf Stream. This structure, designed by the University of Massachusetts (Amherst), is composed of two attached catamaran pressure hulls, 85 ft. in diameter and 800 ft. in length. It is constructed using 2 ft. thick concrete bulkheads, spanned by 5 rows of 3 crosswise mounted heat exchangers, with a 1500 ft. aluminum cold water pipe compartment structure attached to the hulls through a swivel joint, and anchored to the ocean bottom with a double ball pinned joint attached to a sunken barge, 300 ft. x 60 ft. Figure 18 is schematic of the Mark II Concept. The internal hull configuration is made up of a series of 25 Mwe power modules self-contained for independent operation. The Mark II contains 16 such modules producing a net power output of 400 Mwe.

There are two main advantages to this design which its proponents had presented. First, the location of the plant in the Gulf Stream hopefully enables it to reduce the warm water pumping requirement drastically enough to offset the added

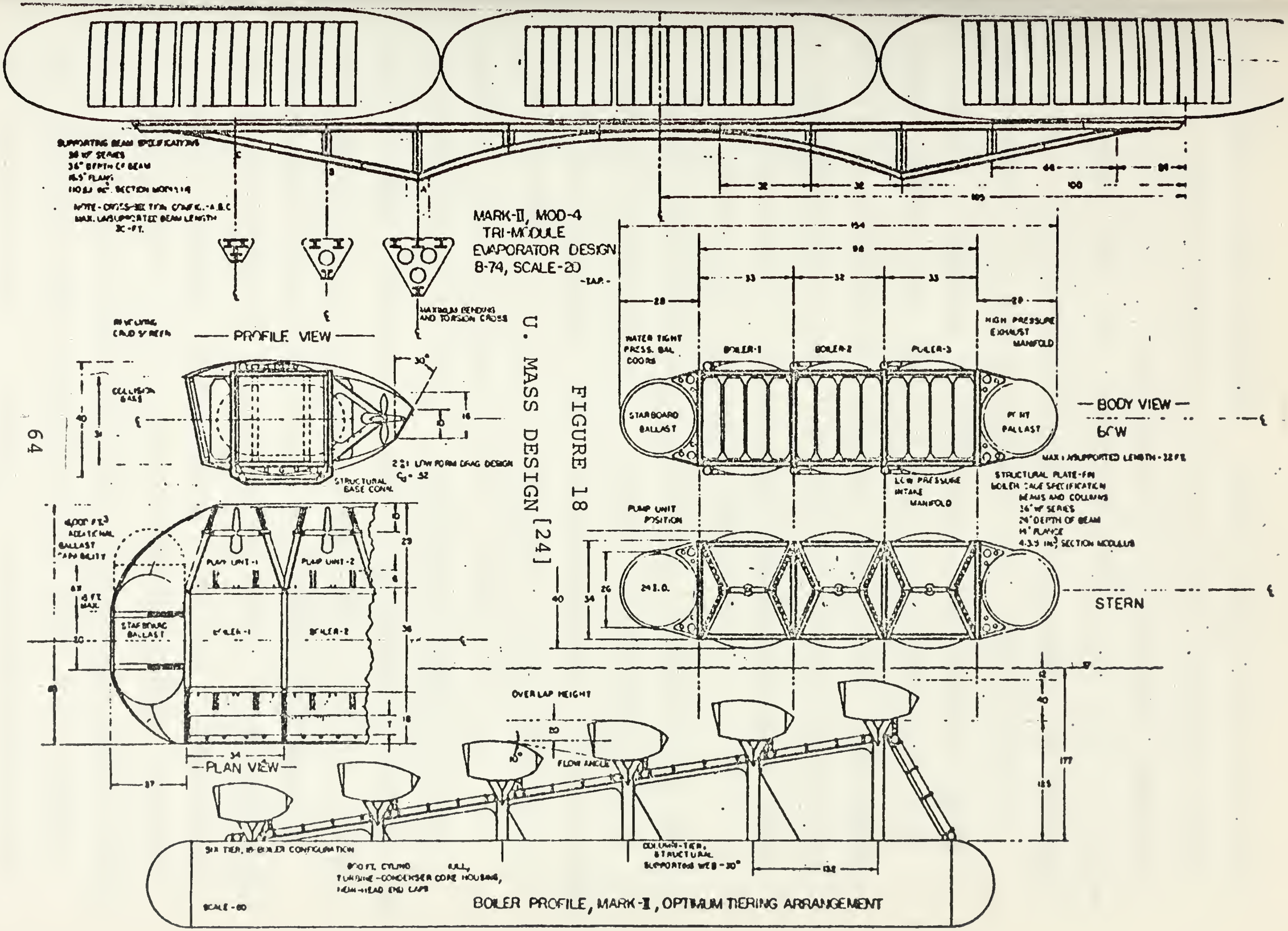



FIGURE 18

U. MASS DESIGN [24]

complexity of the structure. Second, such a site within the Stream, would be relatively close to the continental United States power grid. After extensive oceanographic research, it was found that such a suitable site existed within 30 miles of Miami. Power transmission technology exists now to enable transmission of electrical energy with a high degree of efficiency. The UMass team had also realized that a large enough resource area of solar energy surrounded this site which could accommodate additional OTEC plants. This added to the advantage of being able to use economics of scale in the installation of a common power transmission grid system for the entire plant generation system.

While these advantages were strong enough to make Gulf Stream siting look attractive, there are some serious offsetting disadvantages. This location entails a lower available $\Delta T$ and therefore lower plant efficiency. Plant cost is related to $\Delta T$ as the function of $\Delta T^{-k}$ where $k = 2$ to 3, so increased costs are accepted. Turbulance generated around the evaporators could cause a loss of adequate warm water flow rates and reduce plant output. Compensation for this problem necessitates the introduction of warm water pumps negating some of the advantages of Gulf Stream siting. The submersible structure is possibly too complex to be built economically. Life support and buoyancy control systems reliability might drive construction costs too high. This system design assumes that electrical power transmission via power cable, should other forms of power or electrical transmission appear more

economically attractive than the design might not be flexible enough for conversion. Finally, the OTEC concept, to be useful, will require numerous plant sitings, approximately 50 of 400 Mwe UMass types to produce 1% of U.S. year 2000 energy needs. To concentrate this number of plants in a limited area of the Gulf Stream could cause a reduction of its temperature resulting in unpredictable climatic changes on the continents bordering the North Atlantic basin.

Because of the complexities involved in the Gulf Stream design, the stagnant pool has been more popular as the design environment. Here the problem of drag forces are eliminated. Designs then concentrate more on such issues as ease of construction, ability to withstand environmental forces, and most efficient plant configuration. In the Gulf Stream at $\Delta T$ of 32°F imposed higher costs than the high of 40°F $\Delta T$ available in tropical seas; and more flexibility in plant positioning methods was now available.

The configurations of the stagnant pool designs have taken two forms: the floating vessel and the spar buoy. The floating vessel uses the standard principles of naval architecture to design a floating hull contained power plant. Figure 19 presents the first floating platform design proposed of J. Hilbert Anderson in 1966. Constructed of steel, it was proposed that this type of plant could be built in existing ship building complexes where work on oil platforms was performed. Some modification to existing facilities might be necessary. The long cold water pipe was attached later at

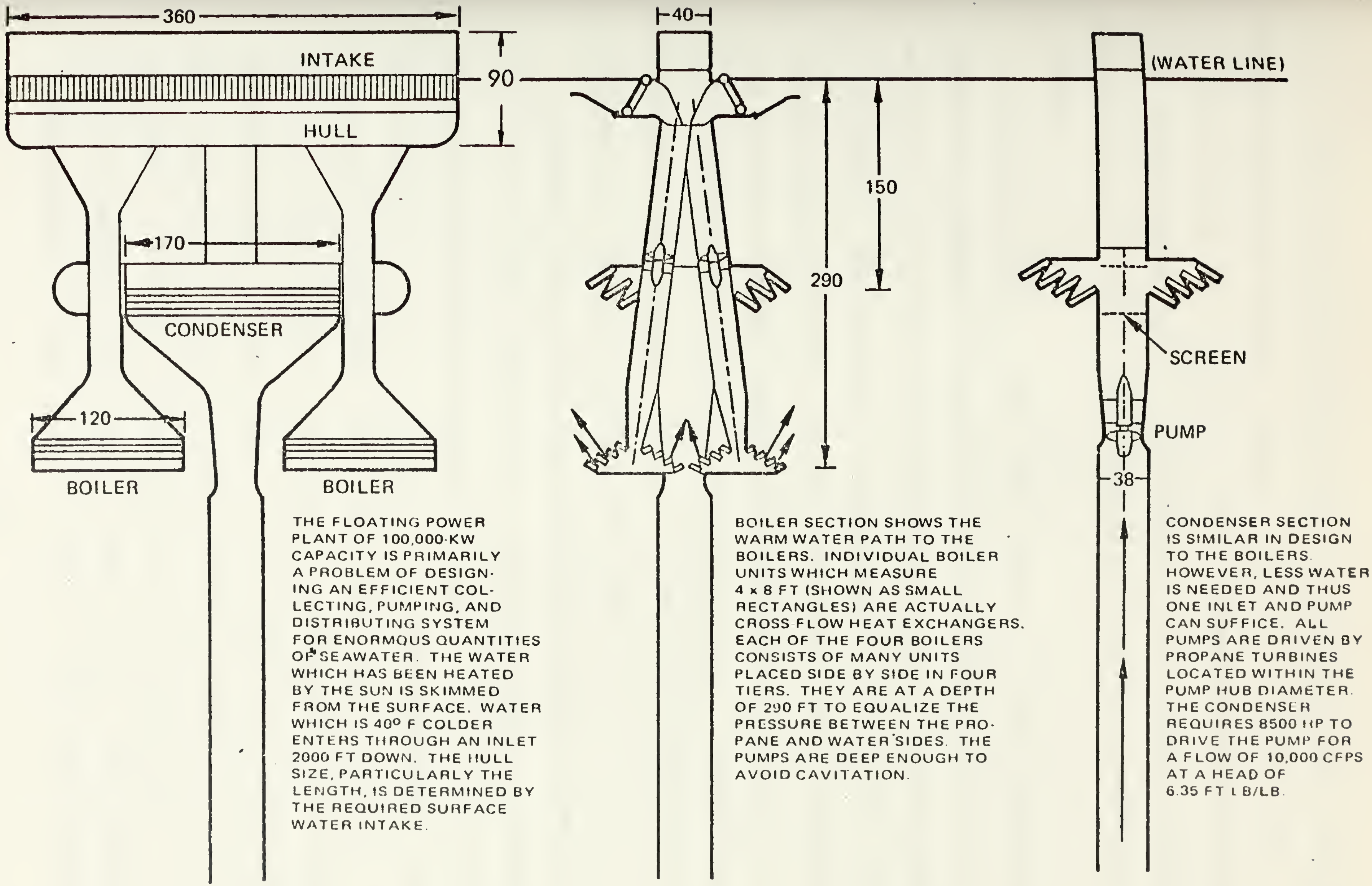



FIGURE 19

ANDERSON OTEC CONCEPT [23]

sea. The dimensions of this plant are noted within the above mentioned illustration. Again, its size is quite large. Notice in this design the location of the condenser above the evaporator taking advantage of the working fluids higher pressure of evaporation to reduce the thickness of the heat exchanger walls.

The floating hull has the advantages of not having the larger and expensive buoyancy control systems of the submersibles yet it does have the disadvantage of being subject to effects of seven weather conditions and collisions, of relying upon some form of dynamic position which can impose high parasitic power demands, and of being subjected to deterioration over time in the splash zones. However, no additional life support systems are necessary other than as required by maritime practice and law.

The spar buoy is usually a neutral buoyant submersible. Figure 20 is such a structure. The power plant is located in the upper part of the structure within the first 500 feet of ocean surface. The hull design is highly functional driven by interior arrangement more than other considerations. The major loading upon the structure is hydrostatic with some effects of wave height added. The problems of storm action and collision have been eliminated due to the depth of the structure. Position is maintained by various types of anchoring methods vice dynamic positioning. The disadvantages of this system are the same as some of those experienced by

the Gulf Stream submersible, i.e., the maintenance of a life support system if manned or an automatic monitoring and control system if unmanned, and the operation of a ballast and deballasting system to insure that the natural buoyancy of stability of the structure remains intact.

The stagnant pool designs have the advantages over the Gulf Stream system in that they offer more flexibility for energy transmission alternatives. Ammonia synthesis or hydrogen production platforms and storage facilities could be more readily attached to such structures. The disadvantages of these structures are that the area in which suitable temperature gradients exist are not located near the continental United States and the deployment of such a plant will occur a significant distance from both construction and repair facilities. As experience has shown in the North Sea, such deployments are very tenuous undertakings requiring long periods of good weather for a very low speed, (2 knots) transit. These distances also render conventional power transmission technology inadequate to provide economical power.

## SUMMARY

As a short term solution to the energy problem, ocean thermal energy conversion does not appear to be a viable alternative. The uncertainities involved in its development and capital costs required for its initial deployment are presently prohibitively high. However, as a possible long

range alternative, the OTEC concept should not be discarded. It has the potential of producing energy from an inexhaustible and free source. This is also pollution free energy so the social penalities received from fossel fuel plants are not anticipated.

Research applied to ocean thermal energy conversion has established that the theoretical foundation for program development does exist. Technological carryovers from other endeavors such as offshore oil exploration are available to exploit in the implementation. Yet serious obsticles do remain, particularly in the heat exchanger design. Presently, they are just too large and therefore too expensive. In some designs, the costs of the heat exchangers alone exceed the total power plant capital cost of other alternatives. OTEC plant design is in the embryonic stage and breakthroughs are necessary for the plants to be economically competitive.

In this chapter, it is attempted to give the reader a perspective of the OTEC plant, its size, shape, and internal components. The contents draw heavily upon university research, piecing information together from various sources. These designs have not been integrated into a preliminary design of a possible OTEC candidate. In what follows, we will discuss and illuminate upon two industrial grouping's attempts to engineer the OTEC plant concept into a baseline design.

# CHAPTER II

# BASELINE DESIGN

## INTRODUCTION

In Chapter I, the Ocean Thermal Energy Conversion plant concept was discussed in an analysis of series of loosely integrated yet separate components of the overall system. This procedure was used as the way in which research has evolved in the direction of optimization of individual system components' design with optimization of the total design being suppressed until the feasibility of the components could be definitively demonstrated. Under these conditions, it is difficult to gain a clear perspective of the OTEC plant system. It is impossible to know where one stands unless a means of comparison is devised. This is the purpose of the baseline.

In 1974, the U.S. Energy Research and Development Administration, assuming the solar program responsibility from the National Science Foundation, contracted two industrial teams, headed by Lockheed Missiles and Space Company, Inc. and PRW System Group, to provide the baseline designs of the Ocean Thermal Energy Conversion plant as a basis for overall system analysis, evaluation, and future planning. In this chapter, the results of the year long studies developed by these industrial teams will be reviewed and illuminated with emphasis directed in the area of the major system components and overall plant configuration. Basic design assumptions will be

discussed in order to provide a reference for analysis. The parameters used to define systems requirements will be illustrated with their resulting design conclusions for the individual components. Identification of problem areas will be undertaken and recommendations for means to their solution discussed. It is the purpose of this discussion to provide the reader with a definitive picture of "where we stand"..

## THE PHILOSOPHY OF THE BASELINE DESIGN

Figure 21 illustrates the system engineering approach to OTEC plant development as adopted by one of the industrial teams, Lockheed Missile and Space Company (LMSC). This systems approach is similar to that developed by TRW for its analysis and is therefore representative of a design approach. In this figure, notice the position of the baseline design within the overall framework. It is the initial focusing point of various component designs into a conceptual design of the entire system. Here, the foundation for the analysis of engineering issues is laid. It becomes a reference point from which the efforts on research and design can be direct culminating in the preliminary design of the OTEC plant.

The initial point of departure in the formulation of a baseline design is the establishment of a system definition. Before starting to consider various alternative design, an entire plant will be defined as minutely and clearly as possible. In OTEC, this involves establishing a hierarchial level of description ranging from the general plant description

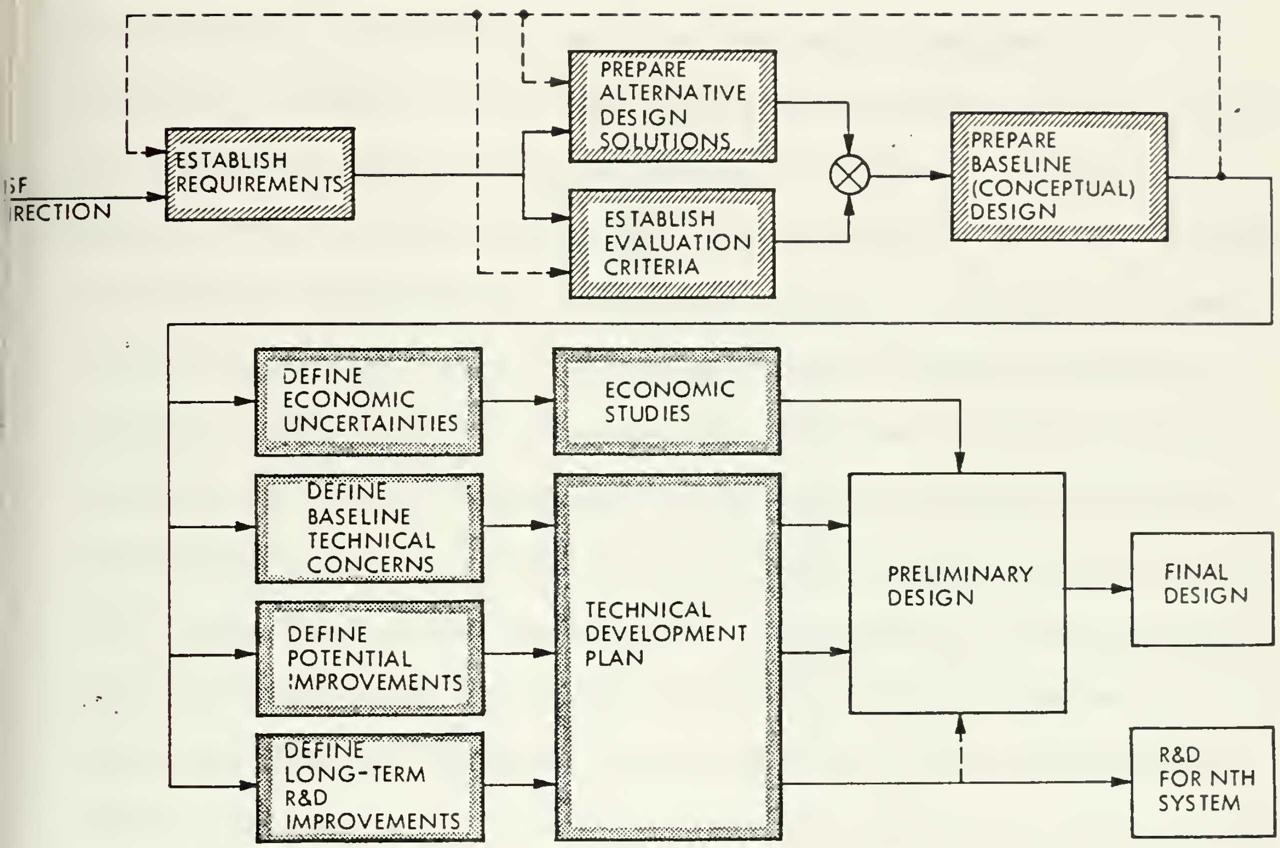



FIGURE 21

SYSTEMS ENGINEERING APPROACH TO OTEC DEVELOPMENT [22]

to various subsystem components; establishing a hierarchial level of performance, physical, reliability, maintainability, availability, system effectiveness characteristics; defining environmental conditions such as sea water temperature, pressure, humidity, wave and hydrostatic loading; and considering the various facets of design and construction. Certain assumptions or ground rules are established to provide a means of comparison with other baseline analysis. OTEC plants were not to be designed as to be constrained to operate within a specific ocean site so that design could be flexible enough to adapt to various requirements of power transmission and proximity to demand as would be defined in later analysis. This leads to a second assumption that baseline design would occur isolated from any consideration of energy transmission. When cost data was produced, it would be in 1974-75 dollars. Technology used within the plant design would be state-of-the-art extrapolation with a minimum of research. The ultimate objective would be to provide 100 $M_{we}$ electrical power to the busbar at minimum cost. Baseline systems would approximate the optimum economically but would not then be considered nor be presumed to be so. The baseline design would be one which embodied the basic system principles with the lowest possible technical risk. Performance, reliability, and operating factors outweighed cost. Where various options were regarded as having equal merit and risk, the lowest costing option was selected. Risk was defined as the technological immaturity and uncertainity of a system.

Upon the establishment of system definition and design assumptions, a review of existing theory and accomplished work is necessary. As we see from Chapter I, the work of various university teams and private groupings had, to a large degree, provided the necessary information to broadly define the nature of the problems involved in an OTEC design. This information was absorbed by TRW and LMSC. It was further enhanced by the addition of current market information on material cost and availability, of contractor experience in component design and fabrication, and, where necessary, of changes in performance criteria resulting from later research work. The information resulting from this step was integrated with system definition to produce the baseline design. From here conclusions could be drawn for the identification of high leverage on risk items and action for required research and development programs to reduce these risks could be initiated.

## OPERATING ENVIRONMENT

The most important environmental factor affecting the design is the available $\Delta T$ in the ocean site. The design will be based upon the worst case $\Delta T$. In the northern hemisphere, the worst case $\Delta T$ will occur when the least amount of solar energy is available for absorbtion by the ocean surface. This will be during the winter months. In both the LMSC and the TRW baseline designs a $\Delta T$ of not less than 34°F during the winter months was considered adequate. The temperature gradient was to occur over a depth not to exceed 3000 feet

yet the warm layer was to be deep enough to permit location of warm water inlets below the surface preventing intake of potential polutants and fouling materials. A sustained surface current in excess of one knot but nor more than two knots was desirable to remove the used effluents. The area of the ocean meeting these requirements must have been of such a magnitude to support numerous OTEC plants (in excess of 100). This last constraint eliminated the Gulf Stream type structure from consideration since this constraint in conjunction with the other constraints became binding.

The location of the OTEC site near the continental United States was adjudged to be desirious. If near enough to a population center, the busbar cost of the OTEC would be more realistic for direct comparison with other power generation alternatives. A grid accessible to plug-in by the OTEC would solve transmission problems. These factors would enhance the desirability of the OTEC, yet were deemed as not overriding. Therefore, the OTEC ocean site selected was to be in the open ocean for both baseline designs. Sites in the southern Carribean and mid-Pacific were considered as acceptable.

LMSC decided upon a submerged spar buoy as the platform for the plant. This decision simplified LMSC concerns to that of assimilating sufficient data to determine available $\Delta T$ and subsurface currents. The plant would be submerged 100 feet and for the most part be subjected to hydrostatic loading only. The effects of surface conditions were minimal;

the proposed plant could maintain operations in severe weather with surface conditions as high as sea state 9 on the Beaufort scale. The TRW design would be a floating cylindrical structure; and the absence of severe storm activity was a criteria in site selection. The design is influenced by more traditional offshore structure constraints such as maximum wave height (100 feet), maximum wind speed (100 knots), and the ability to withstand large impact loadings. For these reasons the LMSC design is more flexible for siting purposes, while the TRW one favors siting in the southern Carribean region.

The site depth is only important in that it be deep enough to permit formation and exploration of a permanent thermal gradient by the plant. The TRW plant is free floating and unaffected by depth. The LMSC spar buoy is designed to be anchored with a trapeze-like cylindrical chain-linked anchor line which can be placed in depths up to 20,000 feet.

## DESIGN CONCLUSIONS

Hull and Plant Configuration - The type of hull conceptualized and configured is based upon an analysis with the following determinants; the cost and use of hull construction, safety and maintainability, station keeping requirements, and versitility of hull design to adapt to various energy transport alternatives. The material to be used for hull construction was selected upon conclusions drawn from a trade-off analysis of the required amount of material necessary to

achieve design strength versus its cost, previous experience with this material in offshore construction, cathode protection requirements of the material in the ocean environment, a plant design life estimate and the susceptability of the material to the fouling agents in the marine environment.

Figure 22 is the hull configuration proposed by the Lockheed team. This structure is a spar bouy with a central core of concrete encased flotation cylinders, massive ducting arrangements for effluent flows, crew quarters, storage, and auxiliary machinery spaces. This central core forms the major sections of what is the baseline platform with other components being the trapeze mooring system and a telescoping cold water pipe. The platform is designed to operate as fully integrated units with four separate power modules attached. Without including the cold water pipe and the power modules, its dimensions are 592 feet high, of which 532 feet will be submerged during operations, and 246 feet outside diameter. The upper flotation cylinders (four of them total) are 277.5 feet long with a 54 foot outside diameter, the lower ones (eight in total, ringing the structure) are 154.5 feet high and 48 feet in diameter. Rising above the central core will be a steel stand pipe housing plant access and the crew quarters. The upper sixty feet of the stand pipe will be the only section of the structure above water. The main platform will float 100 feet below the surface so as to be free from the effects of the ocean surface conditions

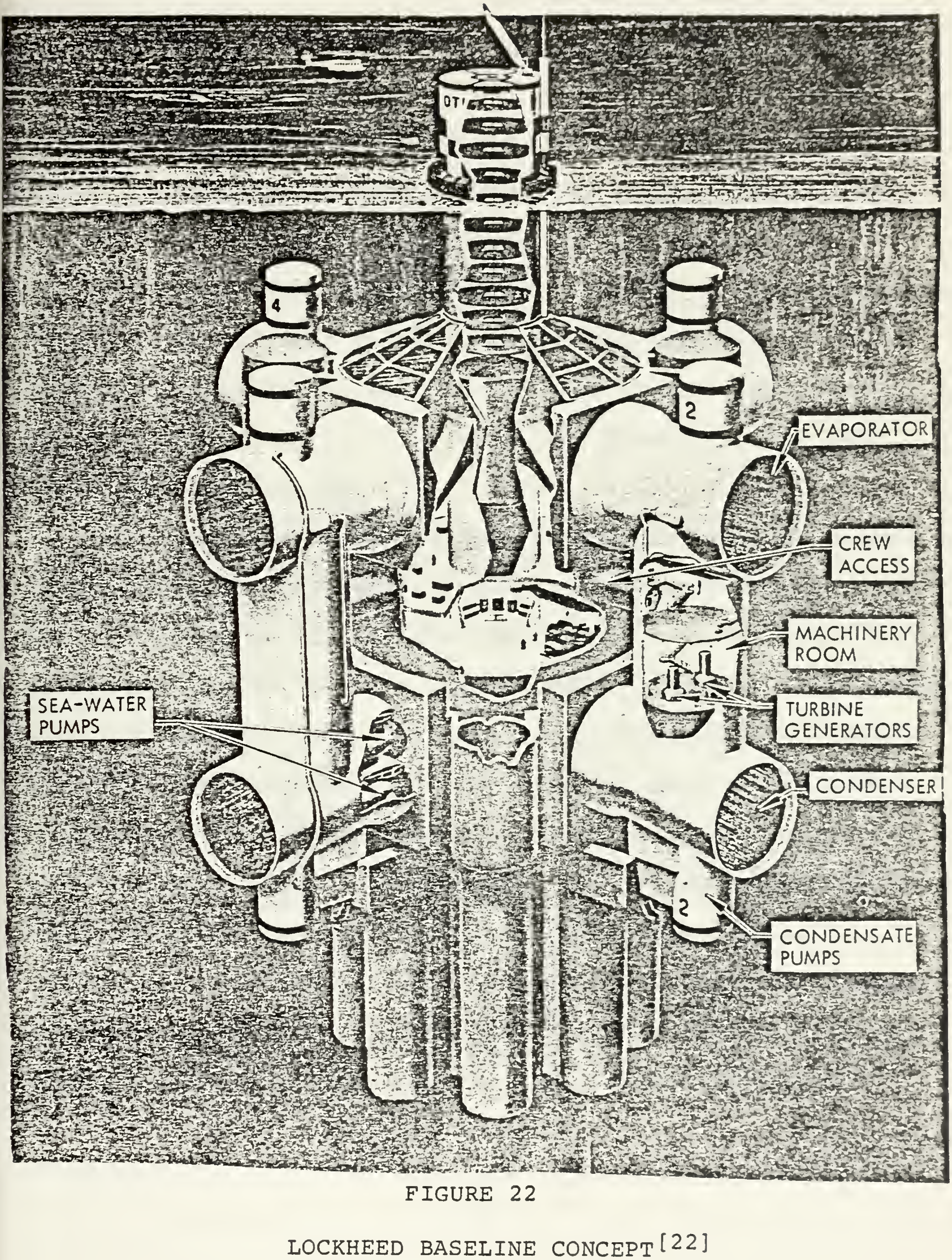



FIGURE 22

LOCKHEED BASELINE CONCEPT[22]

during severe weather and the dangers of collisions with surface vessels. Attached to the lower end of the platform by a joint similar to that of the joint of a telescope is a 1000 foot cold water pipe. It is itself a telescoping structure composed of five 200-foot sections of decreasing outside diameter. The cold water pipe wall is made of concrete, prestressed with internal voids to conserve material. This wall is 1.5 feet thick. The outside diameter of the cold water pipe decreases from 129 feet at the upper connection point with the hull to 105 feet at the lower lip 1500 feet below the ocean surface.

For the plant to be reliable with power packages "easy" to maintain, it was decided to design independent power modulesjointed externally for attachment at installation and detachment for servicing. These modules, rated at 60 Mw gross capacity each, contained internally all the necessary components for power production, i.e., turbine, generators, heat exchangers, and effluent pumps for drawing water through the core platform. These power modules are 332 feet high with a width of 79 feet across the water inlets, the modules' maximum breath. The hull of the modules are fabricated using steel having the advantage of being built and overhauled in a shipyard and being stowed to the platform for detachment.

The conclusions of the TRW team differed from that of LMSC in that it was concluded that a floating plant represented a more feasible alternative than did the spar buoy. TRW estimated that the factors of previous experience, insurance rates, manning, and constraints would weigh more heavily in favor of a floating platform than the submersible or semi-submersible. The risks involved with a spar buoy exceeded those of the floating platform. To TRW, a floating cylindrical vessel appears optimal under the design parameters.

Figure 23 is their proposed structure. It presents a floating cylindrical concrete hull with dimensions of 340 feet in diameter and 170 feet in height. When deployed, excluding the extension of its cold water suction pupe, it will have a draft of 110 feet. While the hull is a monolithic structure, the plant is internally modularized (Figure 23a) into four independent plants with a 32 Mu gross power output per module. This is to benefit from the optimal size of the 25 Mw net output turbine increased reliability of plant operation, and smaller, more feasible sizing of plant components. Each unit contains its own evaporator, condenser, pump and turbogenerator. The cold water pipe, unlike the Lockheed design, is light weight tubes made of fibre glass reinforced with steel rings and rods to preserve strength yet the pipe remains flexible. It is attached to the hull using a hemispherical bearing placed upon the end of the pipe as it is lowered by crane through the center of the cylinder during deployment.

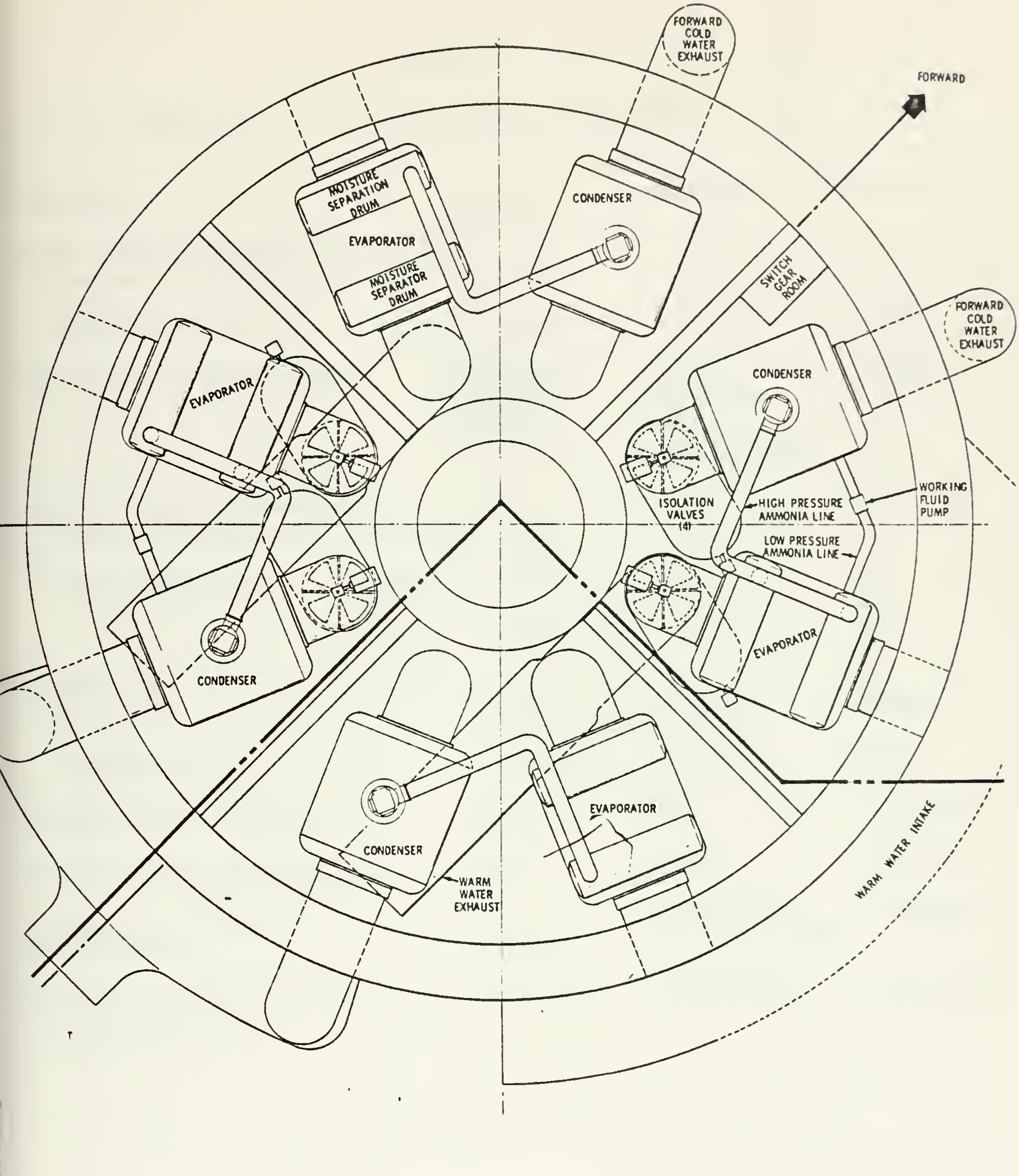



FIGURE 23a

PLAIN VIEW OF TRW BASELINE[23]

TABLE 4

COMPARISON OF WORKING FLUID CANDIDATES[23]

| CRITERIA | | AMMONIA | PROPANE | R-12/31 |
|---|---|---|---|---|
| Heat Transfer Characteristics | | | | |
| Thermal Conductivity | Liquid | .29 | .07 | .05 |
| (BUT/hr-ft$^2$-°F) | Vapor | .014 | .01 | .006 |
| Material Compatibility | | not with copper alloys | some plastics not compatible | inert |
| Termodynamic Properties | | | | |
| Heat of Evaporation (BTU/lb) | | 500 | 140 | 70 |
| Heat Capacity | Liquid | 1.13 | .62 | .24 |
| | Vapor | .019 | .24 | .097 |
| Safety | | | | |
| Toxicity | | severe | slight | slight |
| Flammability | | moderate | severe | none |
| Solubility in Water | | high | low | very low |
| Supply Availability | | good | good | limited |
| Effect on External Environment | | slight | undesire-able locally | unknown |
| Effect of Contamination of Working Fluid | | moderate | negligible | negligible |

Like the LMSC concept, the cold water pipe provides common ducting for the cold water intakes for each module. Unlike it, warm water intakes are located on the outer surface of the modules and do not draw from a common duct.

The TRW system is a floating plant and unlike the Lockheed spar buoy this system is not attached to a static anchor. The effluents discharged are used to position the plant dynamically. In this system a beacon or guide wire is attached to the ocean bottom at the desired site. As the plant moves away, pushed by environmental forces, a computer using the beacon or guide wire as a reference causes a water jet system to activate moving the plant back on station. A dynamic positioning system has the advantage of eliminating most constraints placed upon a plant siting by depth like high cost and weight of traditional anchoring systems. It has the disadvantages of requiring a high initial capital cost and can impose a heavy parasitic load upon plant output. TRW concluded that a dynamic positioning system would be a highly reliable system requiring little maintenance with a design life far longer than a static anchor system. Given these advantages, a dynamic positioning system was proposed.

Working Fluid Selection - In the previous chapter, it was shown how comprehensive was the work done by the academic research teams on the problems involved in working fluid selection.

The conclusions arrived at by the industrial teams were heavily influenced by this research work. TRW based its selection upon the overall trade study of a working fluid's heat transfer characteristics, materials compatibility, thermodynamic properties, safety, soluability in water, supply availability, effects upon external environment, and the effects of sea water contaimination upon the working fluid. Lockheed looked at the same criteria and added a final criteria called effects upon design point. This added criteria gave the decision the dimension of guaging spillonew into other design areas.

Both teams concluded that ammonia would be the most desirable. Table 4 is a comparison of working fluids based upon the TRW criteria. Table 5 illustrates the trade off of heat exchanger material and working fluid. Some of the problems associated with ammonia as a working fluid along with suggested engineering solutions appear in Table 6. Finally, Table 7 presents a comparison of working fluids (ammonia vs. propane) as they effect design parameters. Note how the selection of propane will result in a system with a significantly increased cost. A perusal of these tables will show why ammonia is the most competitive of the working fluid alternatives.

<u>Heat Exchangers</u> - Early research in an OTEC plant design porgnantly demonstrated the costly nature of the heat exchangers. The relationship of 50% of total capital cost was found to be relatively constant for all designs. This

TABLE 4

COMPARISON OF WORKING FLUID CANDIDATES[23]

| CRITERIA | | AMMONIA | PROPANE | R-12/31 |
|---|---|---|---|---|
| Heat Transfer Characteristics | | | | |
| Thermal Conductivity | Liquid | .29 | .07 | .05 |
| (BUT/hr-ft$^2$-°F) | Vapor | .014 | .01 | .006 |
| Material Compatibility | | not with copper alloys | some plastics not compatible | inert |
| Termodynamic Properties | | | | |
| Heat of Evaporation (BTU/lb) | | 500 | 140 | 70 |
| Heat Capacity | Liquid | 1.13 | .62 | .24 |
| | Vapor | .019 | .24 | .097 |
| Safety | | | | |
| Toxicity | | severe | slight | slight |
| Flammability | | moderate | severe | none |
| Solubility in Water | | high | low | very low |
| Supply Availability | | good | good | limited |
| Effect on External Environment | | slight | undesire-able locally | unknown |
| Effect of Contamination of Working Fluid | | moderate | negligible | negligible |

TABLE 5

TRADE-OFF OF HEAT EXCHANGER MATERIAL AND WORKING FLUIDS[23]
(in millions of dollars)

| COMPONENT | Titanium / Ammonia | CuNi / R-12/31 | CuNi / Propane |
|---|---|---|---|
| Heat Exchanger | 80 | 83 | 83 |
| Turbine | 3.6 | 6.1 | 4.5 |
| Piping and Plumbing | 5.2 | 8.5 | 6.1 |
| Working Fluid | 1.0 | 5.3 | .9 |
| Effect of Engine Size on Baseline Hull Cost | .0 | 8.0 | 6.0 |
| Total for 100 $M_{we}$ Plant Size | 89.8 | 110.9 | 100.5 |

TABLE 6

AMMONIA PROBLEMS AND SUGGESTED SOLUTIONS (TRW DEVELOPED)

| PROBLEM | SOLUTION |
| --- | --- |
| Corrosivity | Material selection |
| Water Contamination | Design pressure differential in heat exchanger in favor of working fluid side |
| Lack of Lubricant Soluability | Engineering design |
| Toxicity, fire hazard | Detection systems design, ventilation, and safety equipment |

TABLE 7

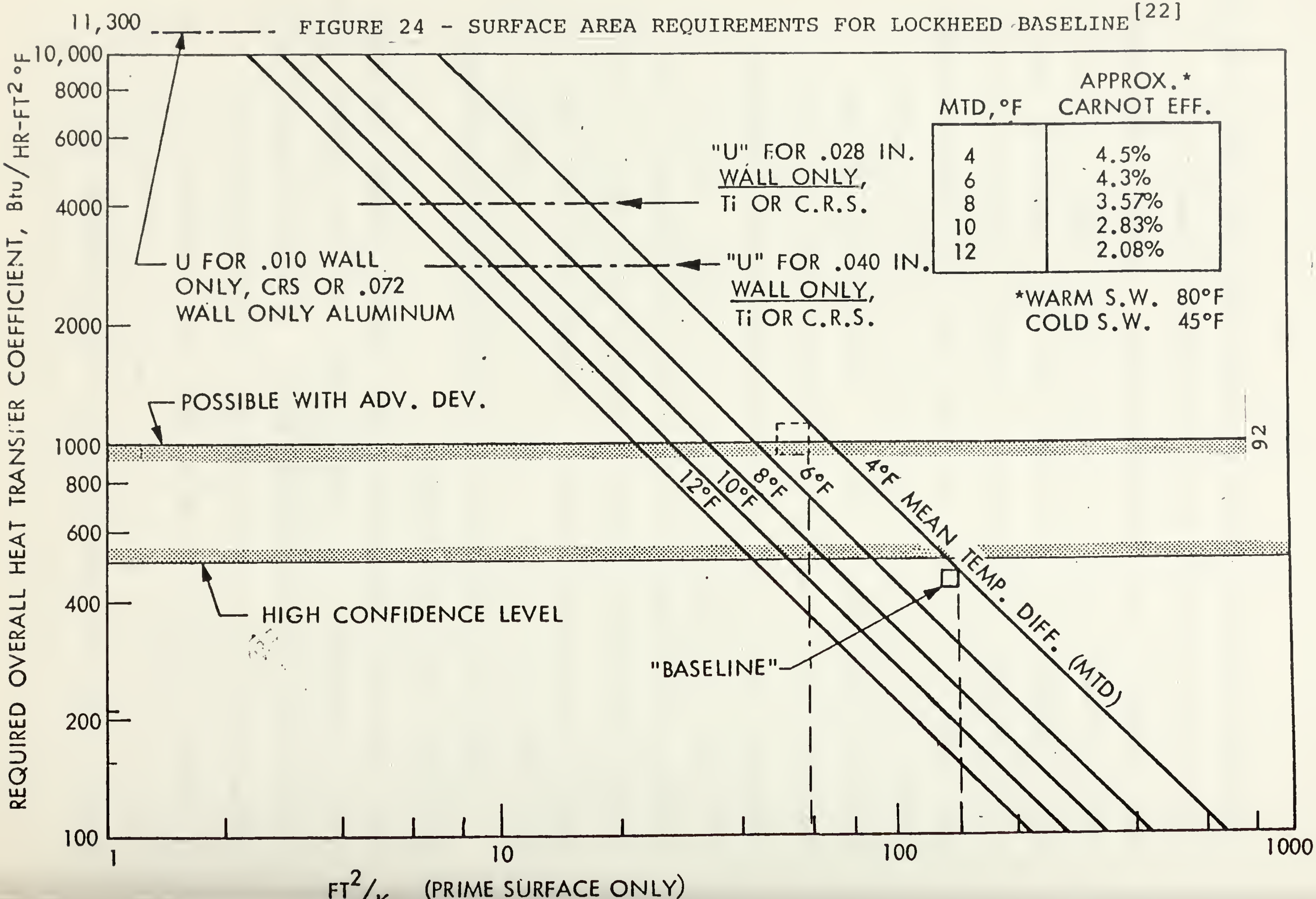
COMPARISON OF AMMONIA VS. PROPANE FOR GIVEN OUTPUT[22]

| System Parameter | Ratio Propane/Ammonia |
|---|---|
| Working Fluid mass flow rate | 3.25 |
| Turbine Diameter | 1.50 |
| Turbine Tip Speed | .55 |
| Turbine R.P.M. | .37 |
| W/F Pipe Diameters | 1.50 |
| Heat Exchanger Volume | 2.50 |

has remained true for the baseline designs. Although this ratio of costs has remained consistent, the estimates of the absolute cost of heat exchange components by the early researchers have been extremely conservative. The early estimate of 40 $ft^2$/kw of heat exchange surface area rose to 140 $ft^2$/kw when estimated by LMSC. Figure 24 is the results of their work in surface area requirements. This variance can be attributed to the optimistic extrapolation of technology by the early research teams.

Previous experience with conventional power plant heat exchangers had strong influence in selecting the heat exchanger fabrication material and configuration. This was a definitive factor in estimating risk, cost, and design life. The limited use of plate-fin heat transfer units coupled with the difficulty in repairing and servicing these vents, all but eliminated the alternative from competition with the standard shell tube configuration (though it might be competitive if fabricated from plastic). Extended heat transfer area alternatives were also considered a well beyond the present technology in fabrication unless used with aluminum surfaces. The standard shell tube type heat exchanger was selected by both teams for the ability to use available technology in fabricating, the ease at which such an exchanger could be repaired, and the relative ease in keeping the tube surfaces clean through mechanical means.

FIGURE 24 - SURFACE AREA REQUIREMENTS FOR LOCKHEED BASELINE [22]



With the selection of $NH_3$ as the working fluid for the thermal cycle, the compatibility of the exchanger surfaces with $NH_3$ is a point of concern. Because of the suseptability of copper to effects of $NH_4OH$ ($NH_3 + H_2O \rightarrow NH_4^+OH^-$ with contamination) the design life of a copper nickle alloy surface would be unacceptably short. Aluminum surfaces could also be suseptable to corrosion under these conditions. Therefore, the major candidates for the material became titanium, stainless steel, and plastic.

Plastic was eliminated from consideration because the only experience with it has been in plate-fin exchangers. The design life of such exchangers is uncertain at this time. Of the two remaining candidates, stainless steel tends to pit in contact with sea water requiring the tube thickness to be twice that as those fabricated from titanium. Titanium, on the other hand, is not affected by the corrosive elements. Although its cost is high, its price has been remarkably stable over the last 15 years. Its production has been limited, but this is a direct function of its low demand. OTEC construction should stimulate this. One plant requires 10% of the annual production of titanium in the U.S. for its heat exchangers. Titanium tube shell heat exchangers with internal pressurization to prevent working fluid contamination were selected by both industrial teams. Tables 8 and 9 represent the final designs by characteristics presented here to illustrate the size of this exchanger and their design variables.

TABLE 8

TRW 100 $M_{we}$ BASELINE EVAPORATOR CRITERIA [23]

| | |
|---|---|
| **OPERATING CONDITIONS** | |
| Number Required | 4 |
| Duty, BTU/hr/unit | $34 \times 10^8$ |
| Seawater Flow, lb/hr/unit | $8.9 \times 10^8$ |
| Inlet Temperature (seawater), °F | 79 |
| Outlet Temperature (seawater), °F | 75 |
| Ammonia Flow, lb/hr/unit | $6.4 \times 10^6$ |
| Ammonia Inlet Temperature, °F | 49 |
| Ammonia Outlet Temperature, °F | 69 |
| Ammonia Pressure, psia | 127 |
| **DESIGN CONDITIONS** | |
| Type | Tube and Shell |
| Design Pressure, psi | 136 |
| Design Temperature, °F | 79 |
| Ammonia Pressure Drop, psi | 15 |
| Seawater Pressure Drop, psi | 2 |
| **CONFIGURATION** | |
| Shell Diameter, feet | 50 |
| Shell Thickness, inches | 1 1/2 |
| Tube Material | Titanium |
| Tube Diameter, inches | 1 1/2 |
| Tube Wall Thickness, inches | .037 |
| Tube Length, feet | 43 |
| Number of Tubes | 75,900 |
| Surface Required, $ft^2$ | 1,220,000 |

TABLE 9

TRW 100 $M_{we}$ BASELINE CONDENSER CRITERIA[23]

| | |
|---|---|
| OPERATING CONDITIONS | |
| Number Required | 4 |
| Duty, BTU/hr/unit | $33 \times 10^8$ |
| Seawater Flow, lb/hr/unit | $7.6 \times 10^8$ |
| Inlet Temperature (seawater), °F | 40 |
| Outlet Temperature (seawater), °F | 45 |
| Ammonia Flow, lb/hr/unit | $6.4 \times 10^6$ |
| Ammonia Temperature, °F | 50 |
| Ammonia Pressure, psia | 89 |
| DESIGN CONDITIONS | |
| Type | Tube and Shell |
| Design Pressure, psi | 115 |
| Design Temperature, °F | 70 |
| Seawater pressure drop, psi | 2 |
| CONFIGURATION | |
| Shell Diameter, feet | 48.5 |
| Shell Thickness, inches | 1 1/4 |
| Tube Material | Titanium |
| Tube Diameter, inches | 1 1/2 |
| Tube Wall Thickness, inches | .032 |
| Number of Tubes | 65,400 |
| Surface required, $ft^2$ | 1,060,000 |

Turbine Design - Having selected ammonia as a working fluid and having been heavily influenced by the theoretical work of Balje, both teams selected a single stage axial flow turbine of approximately 30 $Mw_e$ rated power output as most feasible. A turbine efficiency of 92% was hypothesized achieveable with a dry vapor. This efficiency would decrease at 2% efficiency for every 1% of saturation of the exhaust vapor. In the OTEC power modules, a 98% vapor quality appears likely and the actual efficiency of the turbine would be decreased to approximately 88%.

In addition to the performance qualification, other parameters influencing design included projected cost, ease of manufacture (also related to cost), handling and removal capability, and sizing constraints.

From a preliminary analysis of industrial manufacturers responses, a 20 to 50 $Mw_e$ range turbine output appeared most probable. Surpassing this range, manufacture would not project a design. It required extrapolation of data far beyond of what was prudent or even reasonable.

The size and performance requirements of these turbines exceed the present capabilities of manufactured hardware. Each turbine will have a rotor diameter of approximately 10 feet and weigh 150 tons. To conform with on shelf generator apparatus, design RPM of the turbine was set at 1800. The cost, excluding development, should exceed $70 per kilowatt. Table 10 lists the design requirements of the TRW

TABLE 10

TRW 25 $M_{we}$ POWER MODULE TURBINE PERFORMANCE REQUIREMENTS[23]

| | |
|---|---|
| Horsepower | 41,700 |
| Working Fluid | $NH_3$ |
| Inlet Temperature | 68.6°F |
| Inlet Pressure | 125.7 psia |
| Outlet Temperature | 50.1°F |
| Outlet Pressure | 89.4 psia |
| Vapor Quality | 98% |

TABLE 11

$NH_3$ VAPOR TURBINE CHARACTERISTICS[23]

| | | |
|---|---|---|
| Shaft Horsepower, hp | 44236 | |
| Shaft Speed, rpm | 1800 | |
| Ammonia Flow Rate, lb/sec | 1938 | |
| Overall efficiency | .896 | |
| Specific Speed | 109 | |
| | TIP | ROOT |
| Rotor Diameter, In. | 112 | 73 |
| Inlet Duct Diameter, In. | 64 | |
| Diffuser Exit Diameter, In. | 168 | |
| Height of Assembly (inc. generator), ft | 38 | |
| Turbine Assembly Weight, lb | $50 \times 10^3$ | |
| Power Output, $M_{we}$ | 32 | |

turbine. Table 11 is the TRW baseline turbine design characteristics.

Pumping Apparatus - The last of the major plant components in the OTEC system are the pumping apparatus, warm water, cold water, and working fluid pump inclusive. Table 12 lists the required flow rates maintained. The development of these pumps poses the least uncertainity of all plant components. The warm and cold water pumps are the larger pumps and of a similar design, i.e., impeller type as shown in Figure 25. Note that these pumps are similar in design and construction to conventional two bladed propellers. Their drives will require a horsepower equivalent to a moderate size merchant ship. The reliability of these pumps and parasitic power draw they imposed remain a source of concern as they must be kept within accessible limits. It is estimated that total power loss per module for pumping will be approximately 5 $Mw_e$, the single greatest loss of power.

## MINOR COMPONENTS

To list and discuss just the major components of an OTEC system without discussing the other minor components would hinder the formulation of a comprehensive understanding of the OTEC plant. A detailed study cannot be undertaken as it would be very cumbersome and distracting from this paper's objective. It will not be undertaken. Reference to the baseline design will be sufficient. These components represent the final step in "engineering out" the system.

TABLE 12

FLOW RATES AND POWER REQUIREMENTS

FOR 25 $M_{we}$ MODULE PUMPS[23]

| | Flow Rate lb/sec | Rate & Power hp |
|---|---|---|
| Warm Water | 248,000 | 3200 |
| Cold Water | 210,000 | 4100 |
| Working Fluid | 1,790 | 700 |

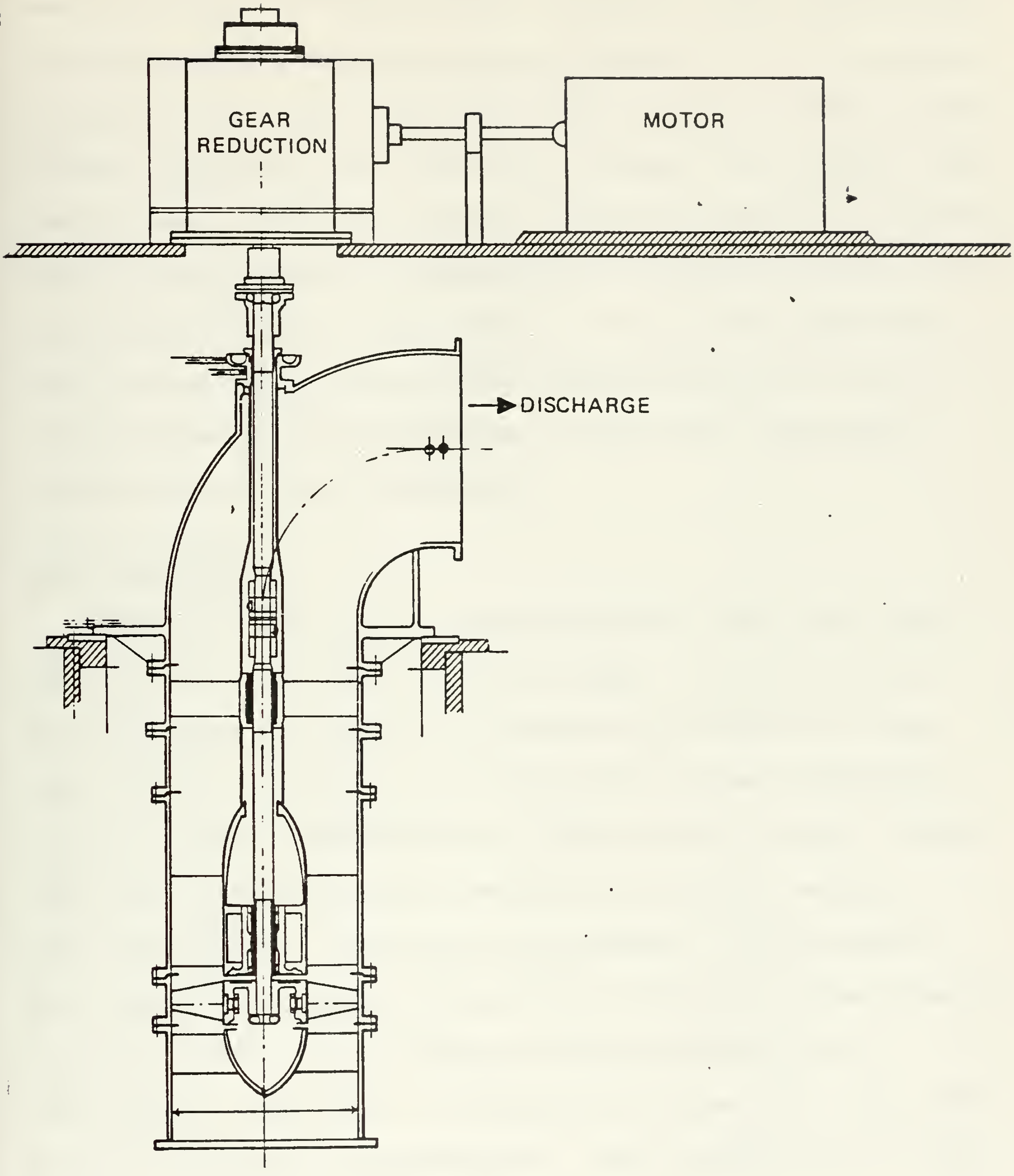



FIGURE 25

IMPELLER TYPE SEAWATER PUMP[23]

They will include various items attached to the system so as to insure compartmentation, proper ducting, and crew access; riggings to the cold water pipe; anti-fouling or chlorination systems; emergency power and start systems; lift-support and damage control systems; and with the plants requiring certification, such systems as mandated by the American Bureau of Shipbuilding and the Coast Guard. The list continues with final evaluation of these items awaiting solution of the issues raised in the development of the major components directly influencing feasibility.

## COST CONCLUSIONS

Tables 13 and 14 are presented here to show the final cost arrived at by the industrial teams. As the range of design assumptions and cost criteria vary between the two teams, direct comparison of costing data is not productive unless a clear understanding of these criteria exist. These costs were assembled from a variety of sources; vendors, trade publications, parametric programming costing models, early proponent's work, and previous contractor experience in projects using somewhat similar design strategies and construction techniques. The range of dispersion among some of the estimates is considerable. The extrapolation of the baseline systems is the cause for this.

The importance of these figures is in the potential magnitude of the cost which they project, in 1974 for development of the OTEC system. They provide the analyist

TABLE 13

TRW BASELINE COST
(constant 1974 dollars)

| Baseline Description (100 $Mw_e$ net) | Unit Cost* |
|---|---|
| Electrical (at busbar) | 7.44 |
| Heat exchanger (8) | 79.83 |
| Cold water pumps (4) | 3.73 |
| Warm water pumps (4) | 2.83 |
| Turbine Generators (4) | 3.65 |
| Ammonia Piping, etc. | 5.22 |
| Ammonia Pumps | .42 |
| Ammonia Inventory | 1.01 |
| Controls power modules | 1.18 |
| Auxiliary system power modules | 5.55 |
| Misc. Power | -- |
| Shell struct. power modules | -- |
| Superstructure misc. | -- |
| Platform structure | 68.79 |
| Mooring System (dynamic) | 1.50 |
| Engineering and Field Start | 23.55 |
| Deployment | .96 |
| Home office services | 4.53 |
| Total | 210.19 |

Capital Cost in $/kw = 2102

*in millions of dollars

TABLE 14

LOCKHEED BASELINE COST[22]

(constant 1974 dollars)

| BASELINE DESCRIPTION (160 $M_{we}$ Net) | EXPECTED MULT. UNIT PRODUCTION COST* |
|---|---|
| Electrical (at bus bar) | 9.2 |
| Heat Exchangers (8) | 236.6 |
| Cold Seawater Pumps (16) | 14.1 |
| Warm Seawater Pumps (16) | 11.9 |
| Turbine Generators (8) | 12.4 |
| Ammonia Piping | 4.7 |
| Ammonia Pumps | .7 |
| Ammonia Inventory | .3 |
| Controls-Power Modules | .2 |
| Auxiliary Systems Power Modules | 6.1 |
| Miscellaneous Structures Power Modules | 8.0 |
| Shell Structure Power Modules | 27.6 |
| Superstructure Misc. | 4.3 |
| Platform Structure | 42.9 |
| Mooring System | 31.0 |
| Deployment | 5.0 |
| Construction Facility Preparation | 5.3 |
| Total | 420.3 |

Capital Cost in $/KW = 2621

*in millions of dollars

with an initial reference for decision making. They identify the high leverage items, for example, the heat exchangers.

The problems with costing data are that they fail to specify the confidence levels associated with the data. They could easily fail to identify a critical item with high leverage. This could become an odious prospect in later development. The minimum expenditure of capital in cost per kilowatt of the TRW baseline is almost three times the cost of a nuclear power plant, three and one-half times that of a coal burning power station, and four times that of an oil plant in 1974. The magnitude of these capital costs indicated that the OTEC system will require a tremendous outlay of resources.

ISSUES RAISED AND PROBLEMS IDENTIFIED

The baseline designs have resolved few issues, rather they have raised more. This was to be anticipated as they were the first cut appraisal of the complete application of the OTEC concept. The two designs were attempts at engineering the concept to for more detail then previously accomplished. Using the vast experience of their own firms plus that of their subcontractors, they were able to assemble a large degree of documentation supporting their design. Yet, as stated in the development plans, the baseline designs were unable to deal with various problems lacking experimental data to support theoretical conclusions. Successful exploration of present technology could not be predicted

in the development of the major components. The detail of the baseline served to identify even more specific problems requiring solution. What has been developed as a result of these studies is better understanding of the magnitude of the obstacles to development of the OTEC concept into an operating plant. These problems can be divided into two general areas; economic and technical.

Economically, the studies indicated that original proponents have underestimated the cost of their proposals and therefore generated a great deal of optimism which was not economically justified. The variations in costs were by a magnitude of ten in some instances. This has appeared to be a result of system definition variation among the researchers. The OTEC concept has appeared to be less attractive then as originally thought, but it could still remain competitive. If the projections of the industrial teams are accurate, OTEC plants at the bus bar might not appear as unattractive when one considers more than just the capital cost per kilowatt generated. Since solar energy is essentially a free and continually renewable resource, there is neither the generating fuel cost nor the opportunity cost associated with using a limited resource. In this instance, the cost of power generation is a far better indicator of economic feasibility then just capital costs alone. In what follows is a series of calculations which give the cost of power generation for five alternatives; nuclear power, coal, oil, OTEC (TRW), and OTEC (LMSC). These calculations are admittedly rough yet

are close enough to give an accurate picture of the importance of fuel costs in determining the cost of power generated.[22]

Cost of power generated is defined as:

$$BCE = \frac{FAC \times C/kw}{F_a \times T} + C_F + C_{OM}$$

BCE = bus bar cost of electricity, mills/kwh

FAC = fixed annual chargerate, percent

C/kw = capital investment, \$/kw

$T$ = hours/year $\times 10^{-3} = 8.76$

$F_a$ = availability factor, % of time

$C_F$ = cost of fuel in mills/kwh

$C_{OM}$ = operation and maintenance cost, mills/kwh

For Nuclear Power:

$C_F$ = 3.42 miles/kwh (based upon estimate of Mπ Solar working group of \$30 per year per kilowatt generated)

FAC = 10% - most probable mean charge

C/kw = \$863/kw (Table 18, <u>Economic Comparison of Base Load Generation Alternatives for New England Electric</u>, A.D. Little, INC., ref ADL/SMSC - 1/75)

$F_a$ = .73 - (Table 13, ADL/SMSC - 1/75)

$C_{OM}$ = 2.27 (Source: pg.87, ADL/SMSC - 1/75)

$$BCE_{\text{nuclear power}} = \frac{.1 \times 863}{.73 \times 8.76} + 3.42 + 2.27$$

$$= 19.18 \text{ mils/kwh}$$

For Coal:

$C_{OM} = 2.5$ (average value of plants with and without $SO_2$ scrubbers)

$C_F = 14.0$ mills/kwh (based upon $30/ton estimate)

C/kw = $625/kw (ADL/SMSC - 1/75)

FAC = 10%

$F_a = .75$ (ADL/SMSC - 1/75)

$$BCE_{\text{coal}} = \frac{.1 \times 625}{.75 \times 8.76} + 14.0 + 2.5$$

$$= 26.01 \text{ mills/kwh}$$

For Oil:

$C_{OM} = 1.49$

$C_F = 17.8$ (MIT Solar Study Group - $12/bbl)

C/kw = $512/kw

FAC = 10%

$F_a = .75$

$$BCE = \frac{.1 \times 512}{.75 \times 8.76} + 17.8 + 1.49$$

$$= 27.08 \text{ mills/kwh}$$

For OTEC (TRW):

$C_{OM} = 1.7$

$C_F = 0$

C/kw = \$2102/kw

FAC = 10%

$F_a = .9$ (contractor assigned)

$$BCE_{OTEC} = \frac{.1 \times 2102}{.9 \times 8.76} + 0 + 1.7$$

$$= 28.36 \text{ mills/kwh}$$

FOR OTEC (LMSC):

$C_{OM} = 1.7$

$C_F = 0$

C/kw = 2627

FAC = 10%

$F_a = .9$ (contractor assigned)

$$BCE_{OTEC} = \frac{.1 \times 2627}{.9 \times 8.76} + 0 + 1.7$$

$$= 35.0 \text{ mills/kwh}$$

The dramatic effect of fuel cost is apparent. While the OTEC power generation method remains the most expensive, it is nearly competitive to fossel fuel alternatives. These alternatives have a power generation cost heavily dependent

upon fuel cost in boal burning plant, 53% of power costs is fuel and in oil plants, 62% is for fuel. These cost alternatives are very sensitive to fuel prices. Cost esculation above these present prices could render an OTEC plant competitive, assuming that the problem of developing an economic means of power transmission can be solved to make "at bus bar costs" a valid means of comparison.

On the technological side of the question, many uncertainities remain. Heat exchangers, consuming 50% of the capital cost, remain perplexing. Their configuration, shell and tube, is not the most efficient for material useage. The plate-fin is better, yet uncertainities of feasible and economic fabrication remain. The heat transfer coefficient, a function of 50 years heat exchange technological development in design, must be improved considerably if a cost reduction is to result. The material requirements for the OTEC heat exchangers are 10% of the total U.S. production of titanium in 1974. Aluminum alternatives have not been demonstrated satisfactorily. These heat exchanger problems will continue to be key detriments in the success of the OTEC plant.

The ocean environment introduces the possibility of biofouling,an extremely worrisome factor in heat transfer. The OTEC plant will be pumping nutrient rich cold water into a zone of stable oxygen rich warm water--an artifical upwelling with profound biological results. If, as it is theorized that, water flows of certain velocities will prevent biofouling,

shut down of a module or all modules due to maintenance of storm conditions could cause rapid growth of marine life within a heat exchanger reducing the heat transfer coefficient, effectively eliminating the possibility of restarting the module or the plant. If effective cleaning alternatives are not available, biofouling could result in module overhaul and costly redeployment.

Effluent discharges have already been shown to cause recirculation of cold water into warm water intakes in the LMSC design. Redesign of platform arrangements is necessary, otherwise plant efficiency would be further reduced below the present 3% which designers can now achieve.

The list of technological problems continues ranging from those introduced by the proposed structure and the long range effects of sea water penetration upon water lightness, reinforcement, and plant displacement, to the performance of the plant in adverse weather conditions.

The baseline designs have been successful in laying bare these issues and have introduced new problems. Technologically, little progress has been made above the work completed by the university researchers. Progress will be achieved when practical application of theory occurs. A more attuned design can be undertaken. Whether this progress will be worth the expenditures required remains to be seen.

SUMMARY

In this chapter, the baseline designs were presented providing a summation of the work with the OTEC system. The baseline designs represent a compilation of university research and practical engineering experience. Although on many detailed points the contractors varied markedly from each other, a pattern of design strategy has appeared. The OTEC plant concept will require the construction and deployment of a large monolithic concrete structure to an ocean site which is a significant distance from the mainland U.S. electrical power grid. These plants will be modular in design to insure a reliability of 90% plant availability which contractors assume necessary for competitiveness. (This exceeds present alternative power generation techniques by nearly 15%.) The internal components for these power modules will be larger than what is in existance under present technology. Their design requires extrapolation of this technology. While theoretically sound, these applications have not been successfully demonstrated. A tremendous potential for critical design disruptions remains. Investigation of these issues is the next step in the development of the concept.

The succeeding chapter will look at one of the high risk areas identified by the baseline design. This will be the design and construction of the concrete hull structure. The strategies for design and construction of concrete sea

structures will be discussed. The baseline proposals in this area will be presented along with their underlying assumptions and possible alternatives will be proposed.

# CHAPTER III

# THE APPLICATION OF DESIGN STRATEGIES AND CONSTRUCTION ALTERNATIVES FOR CONCRETE SEA STRUCTURES TO THE OTEC SYSTEM

## INTRODUCTION

In the previous chapters, the OTEC system concept was developed from its inception through the latest of its proposed applications in the baseline designs as developed by Lockheed Missile and Space Company and TRW, Inc. This was done to provide a view of the full scope of the developing concept and the implications of its application.

In this chapter one of these implications, the design and construction of a 250,000 ton concrete sea structure, will be developed, by far one of the more challenging areas of engineering practice. The philosophy behind the design and the two most feasible methods of construction will be presented along with the proposed methods of construction. This should raise some issues which must be resolved before the questions of constructability can be resolved.

## DESIGN PHILOSOPHY OF OFFSHORE CONCRETE STRUCTURES

Design of Structures in the Operating Environment - The first of the three phases of offshore structural design is the determination of requirements for platforms performance in its operating environment. The platform in this environment will be subjected to loading from three sources: its own dead

weight, loads imposed upon the structure due to the nature of its operations, and environmentally induced loads. In OTEC, the platform will be required to support the loads induced by its own hull structural weight and superstructure. These are the dead loads. The live or imposed loads become the effects upon the structure due to the weight of its internal storage, plant machinery, power modules, operating crew, vessel servicing, and anchoring connections. The effects of these forces can be readily determined and designed for as there is ample experience in structural engineering with this type of analysis. Environmentally induced loading are a different matter as the ocean environment produces both complex and highly random loading. These environmental loads are the resultants from wind, waves, depth, current, structural permeability, and marine growth.

In the TRW design, the results of wave action could be the most important. To determine the magnitude and intensity of wave action, a great deal of observation of the proposed site will be necessary. Using the statistics from these observations, an idea of the wave activity can be formulated. These forces (the waves induced) are the most perplexing as they are random and highly complex. Analysis of structural response under wave loading is difficult and at best imprecise. Two analytic tools generally available are quasistatic and hydrodynamic field equation analysis. In quasistatic, a model of scale is designed and loaded under similar conditions to the proposed site. Data is taken and analysis accomplished

for the entire structure using finite element techniques. Methods using hydrodynamic field equations predict the boundary conditions about the structure. These methods are difficult for a designer to use and require extensive use of a computer to produce meaningful results. Empirical results have found both these methods conservative in prediction and therefore high margins of safety are required. In concrete design, the problems of stress reversal and reinforcement placement are extremely important. Wave forces will also introduce the problems of impact loading and cyclical loading and unloading. Here experience from breakwater performance will be applicable in assisting analysis.

Next in importance will be the effects of long term static loading upon a concrete structure. Since under the proper conditions concrete is known to fail when exposed to long term static loading, the spar buoy concept of Lockheed might face problems in its latter design life. One method of prediction of time to failure of a concrete structure under hydrostatic loading based upon the formula:[44]

$$\log_{10} T = 25.2 - 27.8\,(P_s/P_{im})$$

where

$P_s/P_{im}$ = ratio of sustained pressure to implosive pressure

$T$ = time in years

Other environmentally induced forces should be of lesser concern except in their combined effects upon buoyancy or stability.

Upon determination of the loads acting on the platform, the task then becomes defining performance specifications of the structure and structural material.

The structural performance of an ocean structure is defined by two limit states, ultimate and service. Ultimate limit state defines under what conditions failure of the structure occurs. Ultimate failure mode is defined as rupture of a major structural member resulting in loss of equilibrium, structural instability and overturning, and implosion. Service limit state defines under what conditions the structure is unable to support its intended operation. This would be excessive roll or pitch resulting in loss of cold water pump suction, structural deformations rendering turbogenerators inoperable, loss of water lightness in the splash due to scouring, and deformation in joints causing loss of water light integrity.

The performance of a structural material is a matter of quality assurance. It defines the ability of the material to maintain its strength and overall durability during the term of the design life. In the ocean environment the requirements of the concrete material are that it maintains its integrity under the various loading conditions, resist chemical deterioration due to sulfact attack, prevent chloride

ion permeation into areas of reinforcement, avoid excessive cracking due to creep, shrinkage and stress, and prevent excessive drag forces from accumulation due to marine growth upon the hull. These problems are ameanable to quality assurance techniques in mix designs and placement. To meet loading requirements, a 28-day compressive strength of 6000 psi is desirable for most of the structures with 9000 psi, 28-day strength concrete being placed in splash zones. To prevent salt water permeation into the concrete, a dense paste with a low water cement ratio is necessary. This ratio should be 0.40. For sulfate resistance, a low $C_3A$ cement is used. The $C_3A$ content should be within the limits of 5 to 8%. Reinforcement and prestress bar and tendons should have the same strengths as normal design techniques with surface of 3" for normal reinforcement and 4" for prestressing tendons. The coarse aggregate utilized must be of high quality, non-reactive and of uniform distribution. Air entrainment of 3 to 5% is required in splash zones. Adequate curing and tight surveillance are paramount to quality control. If the quality of the mix is assured, deterioration of concrete should not be of concern although electrochemical corrosion of reinforcement could be a potential problem.

One final design requirement is necessary for any ocean sited plant--the ability to withstand a large impact load due to a collision with another ship or vessel. The design

must contain adequate structural ductility to prevent catastrophic failure and adequate compartmentation to insure continued stability.

Design of Structure for Deployment - The design for the deployment of a structure for ocean siting is somewhat simplier than that of construction or operation as the time frame for deployment is short. Time dependent loads and deteriorations will not become key considerations unless plant redeployment occurs late in design life. During deployment, naval architectural considerations become paramount. The key issues are stability, structural integrity, drag, draft, dynamic response of the cold water pipe, and ease of maneuverability. The main objective of the design is to minimize the risk and difficulty of the tow and deployment process. The loads encountered will be those imposed by deployment operation and by the environment. Assuming that deployment will occur during periods of excellent weather, a critical assumption due to the complex nature of the proposed OTEC plants, the major loading upon the structure will be normal environmental loads from wave and hydrostatic pressure plus those imposed by the function of deployment. Of the two, the functional loads will be paramount. Stress concentration can be expected in areas of tow line attachment. Drag forces due to vessel configuration and marine growth must be minimized through external hull shape and concrete mix design. High overturning moments should be anticipated in the event of a cable parting during tow. The structure should be inherently very stable.

The draft of a designed structure is important as the structure must be accessible to its ocean site. Bottom contours cannot be forgotten. This might require additional flotation devices to be added to the structure or necessitate rotation of the structure during tow. Stress reversal could occur during rotation and must be anticipated.

The final stages of deployment will occur in an area with no protection from environmental effects. At this stage of deployment simplicity of hull shape and ease of module attachment should be heavily weighed. Adverse environmental conditions could jeopardize the whole operation. Detachment of tow and other procedures necessary for installation are to be done quickly and simply. One can seriously wonder about the attachment of a power module the size of a ship hull using cranes and divers as being desirious at this critical stage.

<u>Construction Considerations in the Design of Offshore Structures</u> - It is the objective of a designer to provide a structure capable of meeting required performance specifications at the minimum of cost. There are many influences which will affect the designer in attaining this objective. The designer will find that his decisions will be functions of already desired performance characteristics for offshore operations and deployment methods. These will be dominant influences but the complete design will also be affected by the chosen area of construction and the methods to be used in the construction of the vessel. The last two influences are by

no means independent of each other but rather interact and are functions of such factors as local topography and industrial capabilities.

The offshore concrete structure represents a challenge to a design because it is a structure built in one environment to be used in another environment possibly of totally different characteristics. In operation, a massive concrete structure will have more competitive features than other feasibly alternatives yet may also be particularly difficult and risky to build if the conditions present at the construction site are unfavorable. Methods of construction will be dependent upon these conditions. If the evaluation of alternative methods of plant construction is to be definitively undertaken, a clearer definition of potential OTEC siting is necessary.

Due to the nature of the size of the OTEC structure, two alternative methods of construction appear most feasible; assembly of a segmented structure or afloat construction of a monolithic structure.

The first method proposed, assembly of a segmented structure, is a process of joining at sea prefabricated subassemblies into the OTEC system. Most fabrication will be completed prior to the arrival of the components onto the site which will be fairly close to shore. The process of construction will go as follows:

a) Prefabrication - using existing shipyard and other prestress concrete pouring sites, sections of the OTEC hull are poured, prestressed, and allowed to cure in a uniform

environment. These facilities should be with easy access to water transport. During prefabrication the contractors will control the quality of the mix with dimensions of the sections produced being within tolerances so as not to inhibit joint sealing later in the construction process. In this later assembly of the prefabricated sections, the exactness of these joints will be critical to the watertight integrity of the hull. One method of exacting the proper tolerances is to pour the sections ajacent to each other using material to separate sections that will not bond to concrete.

As enough sections become available they are joined using tendons (post tensioned) into subassemblies which are themselves post tensioned into even larger subassemblies. The objective here is to fabricate as large a plant member as possible ashore in a less risky environment than at sea. It is necessary that these members be seaworthy themselves being launched upon completion. These procedures for prefabrication similar to those used in concrete barge construction. This is represented in Figure 26.

One alternative to prestressing in a shipyard or other land facility would be to precast the units on barges using slip form or panel from techniques. The barges could be tied alongside industrial wharves with good transport access and where equipment to support the construction could be installed. Block sections with a horizontal cross section of 180 feet by 60 feet have been formed to a height of 30 feet and have been

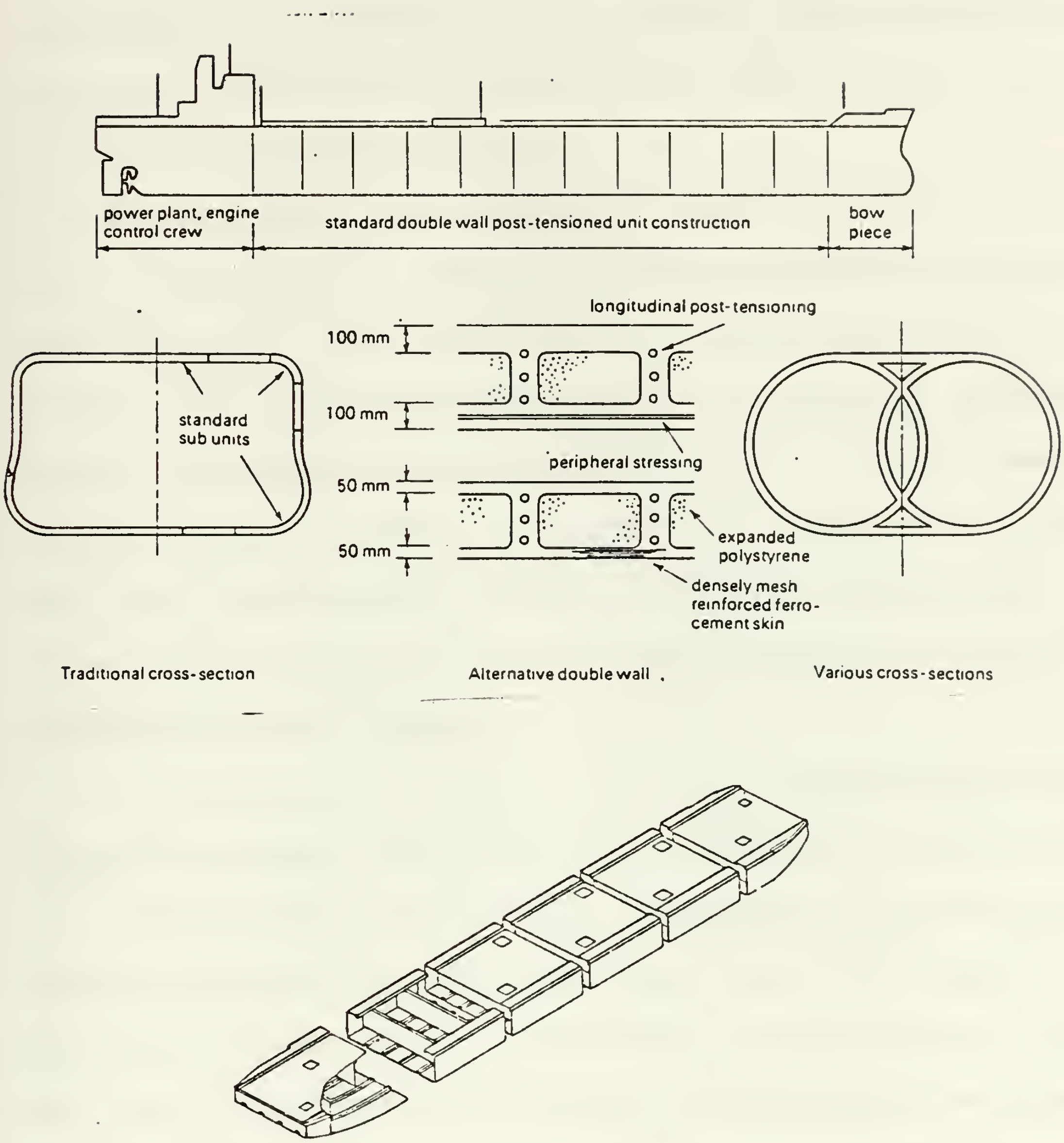



FIGURE 26

PREFABRICATED CONCRETE HULL CONSTRUCTION [26]

precast using this method. There is a serious drawback in opting for this alternative as jointing could be difficult. It would be prohibitively expensive to insure proper tolerance of joints for watertightness.

b) Launching - the method of launching these subassemblies would depend and to a degree determine when the structures would be built. If construction occurred in a dry dock, graving yard, or construction pond, the means of launching would be through controlled flooding of the docks, removal of the retaining cofferdam or opening a flood gate in the dry dock, and floating the section out of the dock for tow. Other alternatives would be launching from shipways or from barges using controlled flooding.

c) Staging and Assembly - as the prefabricated structural components become available upon launching, they can be towed to a staging area in protective anchorage or mooring to await commencement of the final assembly of the OTEC system. This will shorten the at-sea construction period and will prevent any major delays in the problems arising during component construction. Final assembly will occur once all the subassemblies are available. Committment of those resources required for at-sea assembly will start then only.

The final assembly of this large a prefabricated structure is by far the most uncertain portion of the construction process. Previous experience is limited to different fields of construction but nevertheless these techniques might be

applicable. These methods will require that the OTEC hull be assembled such that the plane of its longest dimension lie parallel to the ocean surface. The OTEC plant hull can be assembled using two alternate methods; external framing and falsework, or free cantilever. The first method, external framing, has been used successfully to join hull sections of steel hulled tankers with displacements in excess of 300,000 diret tons. External frames are used to hold the sections in place preventing differential movement while joint bonding and tensioning can be completed. These frames are later removed after assembly so as not to create drag during towing. In the free cantilever method, stays strong enough to support each succeeding attached member protrude from each component. Joining is a process of aligning, contacting, stringing tendons through precast capularies, and post-tensioning. This method is used extensively in prefabricated bridge construction.

Once assembly of the OTEC plant hull has been completed, the plant is towed to its intended operating site and up-ended into position by flooding and ballast adjustment.

Although, as mentioned previously, construction using prefabrication involves many uncertainities, there are several advantages to these methods which deserve discussion. They are as follows:

a) Use of prestressed concrete and post tension technique - By far the most revolutionary method in concrete construction, prestressing has enabled the designer to utilize the compressive

strength of concrete with the added capability of good performance in tension. Prestressing also results in the reduction of material requirements and will reduce the displacements of such large structures as OTEC plants. Prestressing technology is well developed and member standardization is possible to decrease cost of plant mass production. Post tensioning will insure the integrity of joints both in their ultimate load carrying capability and water tightness.

b) Quality control of concrete - Since the quality of the concrete will be a strong determinant of the performance of the hull in the ocean environment, quality control is important to insure the plant achieves design life. Prefabrication will permit easy and close monitoring of the concrete through all stages of construction from mix to pouring to curing. Uniformity of mix quality will be easier and rejection of mix will not be as detrimental to construction time tables as would be during at-sea slip forming. The environment of construction will not be as hostile to the fresh mix as the salt water environment.

c) Delays during prefabrication not as critical as during at-sea slip forming - If standardization of members can be achieved, delays during prefabrication would not be as critical as would be those occurring during afloat construction. Barge, derek, and tug fees are very expensive and lost time during at-sea construction could be far more costly than those occurring during land based prefabrication.

d) At-sea construction time minimized - To anyone experienced with ocean operations, the hostility of this environment to construction operations is well known. Even in moderate seas, crane and derek operations would be both dangerous and difficult. Delay due to weather must be expected. For the long at-sea construction period of an OTEC plant, the risk of severe storms occurring with commensurate damage being inflicted grows significantly enough to worry any underwriter. This time should be minimized. In prefabrication, most of the work will be completed ashore with at-sea time limited to assembly only.

e) Assembly in shallow water - Prefabrication enables assembly to occur in shallower water than would be possible in afloat slip forming of a monolithic structure. In areas of the continental U.S. with an extensive continental shelf extending far out to sea, inshore assembly would reduce transportation distances for men and material to the assembly area.

f) Previous experience at the subassembly level - There exists extensive experience in the U.S. in the construction of barges and dry docks in the fabrication and assembly of prestressed concrete vessels. The problems of concrete in the aqueas environment are not unknown to firms specializing in this type of construction. There also exists similar experience in the bridge construction industry with the design and use of concrete prefabrication.

g) Simplified hull design - In order to optimize prefabricated construction methods, a simple hull design must be undertaken. This will ease the designers task in making structural analysis to determine performance during operation more exact.

Inspite of these strong advantages there does exist some possible debilitating disadvantages which must be reckoned with before prefabrication could be considered completely feasible. These disadvantages being as follows:

a) Lack of previous experience in this large a structure - As of now no attempt to assemble this large a structure at sea has even been attempted. While experience in the subassembly level is extensive and transfer of techniques from shipbuilding and bridge assembly methods possible, this method represents a great deal of extrapolation of techniques. Given the topography of the U.S. east and gulf coasts plus existing industrial capabilities, it might be no more an extrapolation than the proposed use of afloat construction methods.

b) Large number of joints required - Prefabrication will require far more joints than will be necessary in a monolithic slip formed structure. These points will necessitate tight quality control to insure adjoining members fit properly. In most cases, the joints will be formed in a hostile environment increasing the difficulty in getting a good watertight bond. The performance of adhesive materials in the aqueous environment remains very uncertain as experience is limited.

Experience has demonstrated that most bonding materials yield a joint far stronger than the adjacent concrete and methods of pouring such joints to minimize construction time are proven as was evident during the Sidney Opera House construction process.

c) Structure susceptable to differential movement - Because of the high number of joints necessitated by this method of construction, this structure will be suseptable to failure due to differential movement between individual subassemblies after the plant becomes operational. The random nature of loadings in the ocean environment, the difficulty in analyzing the effects of these loadings upon the structure, and the lack of understanding of joint performance during dynamic loading increases the seriousness of this problem.

d) Shallow versus deep water performance - Prefabrication is, by this method of construction, a shallow water method. As such the performance of vessel optimized for shallow water might not be optimal for deeper water. Prefabrication could produce a structure which is suboptimized and therefore be a constraint upon overall performance.

e) External supports might be required - Should the jointings as described prove inadequate for structural performance external supports might be required. The disadvantages caused by these supports would be; the increased use of steel and therefore limiting the advantage of concrete construction, inducement of drag forces during tow, and potential difference in design life from that of the hull.

The second alternative construction method, afloat construction of a monolithic platform, uses a method similar to that performed successfully in the fabrication of North Sea concrete platforms. This method will be discussed in its full detail during the succeeding passages on proposed construction procedures for baseline systems. In order to reduce redundancy they will not be discussed in detail here. Essentially, the afloat construction method is as follows: construction of the base of the desired structure as a raft in a construction pond, graving dock, or barge, flotation of this raft out of the construction area to a site deep enough to permit the draft of the completed structure, raising of structural side walls using slip form operations and controlled flooding of the structure to maintain freeboard at a workable level and to insure overall structural stability and, upon completion of structure, deployment to desired operations area.

The advantages of this technique are as follows:

a) Proven techniques - There has been numerous structures constructed using this method. Construction has been successful and the method has proven itself. There is an extremely large pool of experience available.

b) Extensive use of slipforming - This enables constructors to place these large volumes of concrete in a relatively short period.

c) Minimization of joints - Using slipforming it is possible to limit the number. This can reduce construction time somewhat and reduce some of the complexities of construction.

d) Structural characteristics - Because the structure will be built in deep water, the configuration of the hull can be designed for optimal performance in deep water.

e) Raft provides power and supporting equipment - During raft construction the necessary power generation equipment and tower cranes can be installed within the raft. The raft will support the construction of the final structure using this equipment. A reduction in barge and crane costs can be realized.

This method is not without its own severe drawbacks which must be considered. These being:

a) Long at-sea construction periods - The estimated time for at-sea construction of OTEC plants is 24 months. This is an exceptionally and possibly unreasonably long period to expect that construction can proceed without experiencing adverse weather. Should a severe storm strike, the possibility of severe damage or loss becomes a real consideration. This is a high risk to accept for such a large capital investment. Even if this severe a situation is remote, the difficulty of construction during moderately rough seas should not be lightly discounted.

b) High hidden costs - Construction afloat has a high hidden cost which has not appeared in the popular literature itemizing the final costs. This hidden cost is the cost of construction of the required graving docks or construction ponds. The costs have not been recognized in that they

represent subsidies to the constructor given by the host government and not charged directly to the platform construction. This cost could increase the first OTEC construction cost by at least 50%. The graving dock and supporting facilities cost approximately $30 million.

c) High degree of coordination required - This operation will be extremely intensive as approximately 500 men will be involved during construction in reinforcement placement, pouring, form set-up and removal, equipment installation, and loading and unloading of supply and support vessels. To insure the whole operation goes smoothly, managerial resources will be utilized to their fullest. This type of construction is highly complex. Smooth flow of work is essential to success.

d) High cost of delay during construction - Once forming begins an extensive amount of resources will be committed to construction. These resources include the use of tugs and work barges. They are very expensive and their use must be minimized. Therefore anything which delays this construction process is very expensive.

e) Transport cost and problems - Using this afloat method of construction the distance of the construction area from the shore is very dependent upon depth. On coast with flat gradients, the distance to the site could be considerable. This will entail high transport costs and lost time on resources involved in transit.

f) Higher labor costs - It is reasonable to suspect that differentials to pay rates will be added to normal wage rates due to the environment of construction. This coupled with the long and intense construction period should be a significant cost in the overall construction cost.

PROPOSED METHODS OF CONSTRUCTION

As will become apparent from what follows, the industrial baseline designs were heavily influenced by the work accomplished in concrete construction of offshore structures in an effort to exploit the North Sea oil finds. In this effort, the Ekofisk and Condeep projects have revolutionized offshore structure design and construction. These projects made successful use of available technology, skilled labor resources and topographical characteristics of the shorelines of the northeastern British Isles and the southwestern coast of Norway to construct and deploy monolithic concrete gravity structures into the North Sea.

TRW designed a cylindrical hull floating structure with a diameter of 340 feet with a height of 180 feet. The dimensions of this structure designed to be a monolithic precluded construction in a shipyard. Neither were there available graving docks nor slipways in the United States capable of handling such a structure. Upon comparison with other proposed or constructed North Sea concrete structures, TRW concluded that its cylindrical hull compared favorably with these projects in both physical deminsions and concrete

material requirements. It therefore assumed that such methods used to construct the North Sea gravity structures could be readily adapted to the construction of its baseline design. Its proposal was formulated accordingly.

Lacking the necessary industrial facilities such as a graving yard for initial platform construction, it is necessary to build what is called a construction pond. This involves building a cofferdam in a relatively flat coastal area in the proximity of water deep enough and with direct access to the open ocean to float without grounding the completed structure. Once the cofferdam is completed, the pond is drained and its bed lowered as necessary below sea level to enable construction of a floating raft. This raft which is the bottom of the entire structure is slipformed reinforced concrete with a minimal of prestressing built on top of a course aggregate bed within the construction pond. This bed prevents any vacuum seals from forming between the raft and the bottom of the pond causing the raft to abruptly and unevenly rise when the pond is flooded. Enough tower cranes are built within the raft during this phase to handle work throughout the rest of construction. As soon as the hull walls are high enough to give adequate freeboard to the structure during the next phase of construction, the pond is flooded and the cofferdam removed. The raft is floated away from the construction pond and towed to the site for the next phase of construction.

This phase entails the anchoring of the raft in an area where the depth of water exceeds the structures final draft. It also entails the most extensive construction period in the hull's fabrication as the hull walls will be raised to their final height, all the internal components; the pumps, heat exchangers, turbo-generators, etc. will be installed, the bulkheads which modulize and compartment the structure will be poured, and, upon disassembly and removal of the tower cranes, the upper deck formed with the working equipment for cargo handling, personnel transfer, and cold water pipe deployment then being installed. This is the most critical and riskiest of all phases during construction. Since extensive work will be going on round the clock, bad weather can cause serious delays, damage, or even result in structural instability with the associated catastrophic loss of the structure. The site must be within a protected area where these environmental effects can be minimized. Constant tensioning upon the anchor chains will be necessary to assist in stabilizing the structure and ballasting will be used to adjust the vessels' depth as the slipformed walls continue to rise.

To maintain the pace of operations necessary to minimize construction time, ready mixed concrete, reinforcing steel, and skilled labor must be readily available in sufficient quantities. This means that the construction site must be in the immediate vicinity of land and extensive use must be made of work barges and similar construction platforms.

Upon completion of this phase, approximately 18 months after the start of construction, the structure will be a donut-like structure with a fifty foot diameter hole in the center. Through this hole will be lowered the cold water pipe when the plant is on station at its operating site.

Although the TRW plant has a dynamic positioning system, the plant will be unable to move itself until full plant operation is initiated. This plant must be towed to its operating position. This will require the use of at least 5 to 10 very powerful tugs to move this mass of 212,000 tons displacement. The speed of deployment will be under 3 knots.

Concurrently with the construction of the concrete hull, the cold water pipe will be fabricated separately of steel reinforced fiber glass. It will be constructed in eighty fifty-foot sections jointed at each end so that each section can be attached in sequence forming a watertight seal while being lowered.

With the deployment of the hull, the cold water pipe will be barged to the site in sections. Using the preinstalled crane and winches on the hull deck, the pipe sections will be loaded on board the structure and lowered through the center void. As each section is lowered it will be jointed with the preceeding section. This process will continue until the entire 4000 feet of pipe is extended. The pipe will be attached to the structure with a ball-like joint resting upon a retaining ring.

The proposed construction and deployment scenario of the TRW baseline plant with the exception of cold pipe installation is similar to that of the Ekofisk storage tank project.

While the hull of the TRW baseline design is a monolithic concrete vessel, the Lockheed, due to the exterior to the platform located power modules, is a conbination steel and concrete plant. The construction of these two structural components will occur in different locations. The steel hulled power modules with displacement and dimensions similar to a ship hull will be built in a shipyard. Building techniques to be used will be the same as those for conventional shipbuilding although modified to account for the unusual configurations. It will be built in a dry dock with components installed as construction proceeds. Upon completion of the module hull it will be floated, then towed to the operating location of the OTEC platform for attachment to the main structure.

The main structure is to be built with concrete and constructed entirely at sea. Unlike the TRW method which uses the platforms' buoyancy to support construction, the Lockheed structure will require construction to proceed using a series of platform floats and flotation tanks to support it. It will be essentially built upon barges.

The proposed scenario for the Lockheed plant construction will proceed as follows. Construction will occur upon the ocean in an area protected from the effects of wind and waves

with depth in excess of 500 feet. It will all be done upon a series of concentric circular floats which will be flooded to depth appropriate to formed height of the structure. The first portion of the platform to be built will be the cold water pipe. Using the center most of the construction floats and around a series of flotation 200 feet high cylindrical tanks with an outside diameter less than the pipes' smallest inside diameter, the five cold water pipe sections will be simultaneously and concentrically slipformed. Since the final height of this form will be 200 feet, to insure stability, the center float will be ballasted and the pipe gradually lowered in depth. Upon completion of the cold water pipe, the forms will be rearranged on the next outer ring of construction floats to conform with the cross section of the lower 154.5 feet. Slipforming will commence again. At the 154.4 foot level this process will repeat itself for the remaining 277.5 feet. During these two forming operations, both internal structural bulkheads and external bouyancy tanks will be poured simultaneously. Outfitting and internal structural installations will occur during the upper stages of slipforming and continue for a period there after. When the concrete platform is completed and capable of buoying up its own weight, all floats with the exception of the floats supporting the weight of the cold water pipe are recovered. The upper part of the anchoring system, a chain of cylindrical pinned links, composed of a bridal, trapeze, spreader, and swivel is attached to the platform.

The deployment of this structure, like that of the TRW design, is accomplished using powerful ocean going tugs proceeding at low speeds. The draft of this structure is now 417 feet.

Upon reaching the plant's operating position, using a crane barge with supporting tugs, a single point mooring system is attached to the upper portion of the anchor line. Once the mooring system is completed, the cold water pipe flotation devices are flooded and the cold water pipe is permitted to slowly telescope to full extension. The power modules are towed to the site and attached through the use of controlled flooding, cranes, and divers. After sinking to its design depth, the plant is now ready for operations and power production. Total elapse time of construction and deployment: 24 months.

An alternative method of construction of the Lockheed plant was proposed by the shipbuilding portion of its own organization. This alternative varied from the one finally submitted in that it proposed construction of a raft within a cofferdam rather than the complete at-sea construction of the baseline. This strategy was based upon two assumptions. First, Lockheed Shipbuilding looked at the project as if it would be building it within its own yard or near to the yard. This meant that the topography of the Puget Sound area, where the yard is sited, would heavily influence the design. Puget

Sound does have depths of a thousand feet within close proximity to land and deep access to the ocean although the depth restriction was approximately 200 feet. This did mean that a method similar to the Norwegian method for construction could be undertaken with less open sea construction occurring. It required construction of a cofferdam for a construction pond which delayed delivery of the first OTEC another 12 months. Total time of construction for the first OTEC would be 36 months with delivery of succeeding OTEC's coming at a rate of each 24 months thereafter. Secondly, Lockheed Shipbuilding, concluding that such construction would require use of 500 to 1000 men on site during most of the construction period, realized that considerable savings could be achieved from the use of shipbuilding workers at an average wage rate of $7.25/hour rather than construction workers at the average rate of $11.25/hour. LSCC estimated the cost of construction of the first OTEC would be $71 million under this method. This was considered a more costly method to the TY Lin estimate of $45 million of the at-sea construction method of the baseline. These cost figures compare favorably with the experience of the North Sea structure such as the tanks developed by Howard Doris for the Frigg Intermediate Platform which was a concrete gravity structure costing approximately $40 million dollars.

The implications of these designs to the development of OTEC are crucial and the assumptions behind them need clarification and discussion. First, these designs were developed with the primary emphasis being placed upon the performance of the structure at its operating site, construction and deployment characteristics assumed a secondary importance. To some degree, this was a result of the baseline contract requirements of designing a plant to function irrespective of site location. Second, proceeding from the first assumption, the capabilities of a certain area to employ construction techniques based upon geographical constraints would not be a limiting factor. Third, deployment costs directly related to distance from construction site would be insignificant to overall construction cost. Fourth, open ocean construction of a monolithic structure with a duration of 24 months could be expected to occur without major disruptions due to wave and wind action. Fifth, prefabrication of concrete was infeasible and prestressing methods were to be minimized due to their high steel usage. Finally, transportation from the coastal staging areas would be relatively inexpensive and these lines of supply could be maintained so as to insure no disruption of construction. These assumptions strongly affect the final cost determinations which could readily double as consideration is given to their implication in constructability and it is forseen even by proponents that these designs must be changed to accommodate this.

## CONCLUSIONS AND RECOMMENDATIONS

Design philosophy of offshore structures has been presented as a means of evaluation of proposals for structure. Of interest has been the designs proposed for OTEC systems. Alternative methods of construction of sea structures have been discussed along with those proposed by the industrial teams. The basic assumptions behind these construction proposals have been hypothesized and some criticism or implications applied. Unfortunately it is difficult to draw conclusions at this embryonic stage of OTEC development but rather it is still a period of time for issue raising. Even criticism at the stage is reduced to issue raising until a clearer definition of potential siting locations become available.

Methods used in construction are very dependent upon the area in which construction takes place. Techniques are developed to utilize available resources, be they local geography, skilled labor, material cost, and engineering expertise. In this vein the techniques which evolved so successfully in the Norwegian fjords could be unsuccessfully applied in the Carribean resulting in tremendous capital cost overruns.

Transportation and support costs must also be considered. The inexpensive cost of cement is not maintained when it is transported. This applies for the other structural components. Design must not only consider required performance specifications but also must be undertaken with cost minimizations.

This would dictate construction as near as possible to OTEC plant deployment areas.

OTEC systems represent unique concrete structural applications to the ocean environment as such the methods used to construct them will use novel techniques just as North Sea construction methods were novel due to the nature of those structures.

Many issues remain to be resolved before a conclusion as to the constructability of the OTEC plant. Some required more basic research in jointing method, material, and durability, structural response under dynamic loading, environmentally imposed loads, and optimal shaping of structural members. Some require determination and review of American contractor capabilities, facilities, barge, derek, and towing costs and capabilities, geographical locations for potential construction, and application of alternative construction strategies such as prefabrication.

# REFERENCES


## OTEC Concept

1. Anderson, J. Hilbert and James H. Anderson, Jr., "Thermal Power from Seawater", Mechanical Engineering, Vol. 88, No. 4, pp. 41-46, April 1966.

2. "Power From Ocean Temperature Difference?", MTS Journal, Vol. 8, No. 7, pp. 9-10, August 1974.

3. Fisher, Arthur, "Energy From the Sea...Part II: Tapping the Reservoir of Solar Heat", Popular Science, pp. 78-81, 92, 123, June 1975.

4. McGowan, J.G., et.al., "Ocean Thermal Difference Power Plant Design", Transactions of the ASME, Paper No. 73-WA/Oct. 5 pp. 1-10.

5. Karig, H.E., "Thermal Power Systems Using Ocean Temperature Gradients as Source of Energy", Transactions of the ASME, Paper No. 72-WA/Oct. 12, pp. 1-7.

6. Lavi, Abraham and Clarence Zener, "Solar Sea Power Plants--Electric Power From the Ocean Thermal Difference", Naval Engineers Journal, Vol. 87, No. 2, pp. 33-46, April 1975.

7. Claude, Georges, "Power From the Tropical Seas", Mechanical Engineering, Vol. 52, No. 12, pp. 1039-1044, December 1930.

8. Walters, Samuel, "Power in the Year 2001, Part 2--Thermal Sea Power", Mechanical Engineering, pp. 21-25, October 1971.

9. McGowan, J.G., et.al. "Variations in Heat Exchanger Design for Ocean Thermal Difference Power Plants", Massachusetts University, Amherst, NSF/RANN/SE/GI/34979/TR/74/4, August 1974. PB-238-572.

10. Heronemus, W.E., "Technical and Economic Feasibility of the Ocean Thermal Differences Process as a Solar-Driven Energy Process", Massachusetts University, Amherst. NSF/RA/N 74160. NSF GI 34979 50 P, 30 Apr 1974, PG-239 374.

11. Anderson, J. Hilbert, "Research Applied to Ocean Sited Power Plants", Massachusetts University, Amherst. NSF RA/N 74 002. NSF GI 34979 70P, 25 Jan 74, PB-228 067.

12. Boot, J.L., "Feasibility Study of a 100 Megawatt Open Cycle Thermal Difference Power Plant", Massachusetts University, Amherst, NSF/RA/N 74 109. NSF GI 34979. TR/74/3. 100P, August 1974, PG-238 571.

13. Zener, Clarence, "Sea Solar Power--First Quarterly Progress Report Covering the Period June 1, 1973 to October 31, 1973", NSF/RANN/SE/GI-39114/PR/73/1, October 11, 1973.

14. Zener, Clarence, "Solar Sea Power--Semi-Annual Progress Report, Covering the Period November 1, 1973 to January 31, 1974", NSF/RANN/SE/GI-39114/PR/74/2, January 25, 1974.

15. Zener, Clarence, "Solar Sea Power--Third Quarterly Progress Report, Covering the Period February 1, 1974 to April 30, 1974", NSF/RANN/SE/GI-39114/PR/74/3, April 30, 1974.

16. Zener, Clarence, "Solar Sea Power--Fifth Quarterly Progress Report, Covering the Period July 1, 1974 to September 30, 1974", NSF/RANN/SE/GI-39114/PR/74/5, October 31, 1974.

17. Lavi, Abraham, "Final Report: Sea Solar Power Project, Covering the Period June 1, 1973 to December 31, 1974", Carnegie-Mellon University, Pittsburgh, NSF/RANN/SE/GI-39114/PR/74/6, January 31, 1975.

18. Reynolds, W.C., <u>Thermodynamics</u> 2nd ed., McGraw-Hill Book Company, New York: 1968.

19. Williams, J. Richard, Solar Energy, Technology and Applications, Ann Arbor Science Publishers, Ann Arbor: 1974.

20. Lavi, Abraham, ed., Proceedings, Solar Sea Power Plant Conference and Workshop, June 27-28, 1973. Sponsored by National Science Foundation, Carnegie-Mellon University, June 1973.

21. Harrenstien, Howard P., ed., Workshop Proceedings, Second Ocean Thermal Energy Conversion Workshop, Sponsored by National Science Foundation, Washington, D. C., September 26-28, 1974.

Baseline Design

22. "Ocean Thermal Energy Conversion [OTEC] Power Plant Technical and Economic Feasibility", Lockheed Missile and Space Company, Inc., NSF/RANN/SE/GI-C937/FR/75/1, LMSC Do. 56566, Vol. I and II, 12 April 1975.

23. "Final Report Ocean Thermal Energy Research on an Engineering Evaluation and Test Program", TRW Systems Group, Redondo Beach, Calif., NSF/RANN/SE/GI-C958/FR/75/1, Vol. 1 thru 5, June 1975.

24. Dugger, Gordon L., Proceedings, Third Workshop on Ocean Thermal Energy Conversion [OTEC], Sponsored by U.S. Energy Research and Development Administration, Houston, Texas, May 8-10, 1975.

Offshore Structures

25. Gerwick, B.C., "Prestressed Concrete Floating Platforms and Submerged Structures", Proceedings of the Sixth Congress, FIP, Prague, 1970. London: 1971, pp. 1-10.

26. Morgan, R.G., "Discussion on 'Prestressed Concrete Floating Platforms and Submerged Structures'", Proceeding of the Sixth Congress, FIP, Prague, 1970. London: 1971, pp. 54-56.

27. New, D.H., "Report of the FIP Commission on Prefabrication", Proceedings of the Sixth Congress, FIP, Prague, 1970. London: 1971.

28. Bernstein, L.B., "KisLogubskaya Tidal Power Plant", Proceedings of the FIP Symposium at Tbilisi, September, 1972: Concrete Sea Structures. London: 1973, pp. 66-71.

29. Levi, F., et.al., "Prestressed Concrete Floating Drydock with a Lifting Capacity of 100,000 tons", Proceedings of the FIP Symposium at Tbilisi, September, 1972. London: 1973, pp. 31-35.

30. Lacroix, Roger, L., "Special Problems in Connection with Underwater Oil Storage Tank of Prestressed Concrete", Proceedings of the FIP Symposium at Tbilisi, September, 1972. London: 1973.

31. Gerwick, Ben C., "Considerations and Problem Areas in Design and Construction of Concrete Structures", Proceedings of the FIP Symposium at Tbilisi, 1972. London: 1973, pp. 129-140.

32. Gerwick, Ben C., "Construction of Large Ocean Structures", Journal of the Construction Division, Proceedings of the ASCE, Vol. 97, No. CO1, March 1971.

33. Gerwick, Ben C., "Concrete Oil Storage Tank Placed on North Sea Floor", Civil Engineering, ASCE, Vol. 43, No. 8, pp. 81-85. August 1973.

34. Millbank, Paul, "Platform Construction--The Offshore Potential", Civil Engineering (London), pp. 38-40, February 1975.

35. "Two-Phase System Builds Offshore Footing on Bance and in Deep Water", Construction Methods and Equipment, Vol. 57, No. 7, pp. 98-99, July 1975.

36. Recommendations for Design of Concrete Sea Structures, Federation Internationale de la Preconstrainte, London.

37. Worth, R.E.J. and N.A. Trenter, "Graving Dock at Nigg Bay for Offshore Structures", Proceedings of the Institution of Civil Engineers, Part 1: Design and Construction, Vol. 58, August 1975.

38. Milto, A.A., "Construction of Prestressed Concrete Vessels", Proceedings of the FIP Symposium at Tbilisi, London: 1973, pp. 202-205.

39. Sintson, G.M., "Experience and Forecasts Concerning Application of Reinforced Concrete Sea Vessels and Floating Structures", Proceedings of the FIP Symposium at Tbilisi, London: 1973, pp. 236-239.

40. Sivertsev, I.N., "Perspectives of Concrete Shipbuilding and the Challenge of Prestressing", Proceedings of the FIP Symposium at Tbilisi, London: 1973 pp. 240-242.

41. Mathew, Bryant, "Behavior of Concrete Exposed to the Sea", Civil Engineering in the Oceans, II, ASCE, December 1969, pp. 987-998.

42. Hove, Knut and Foss, Ivan, "Quality Assurance for Offshore Concrete Gravity Structures", 1974 Offshore Technology Conference, Houston, Texas, Vol. 11, OTC 1948, pp. 829-842.

43. Haynes, Harvey H., "Long Term Deep-Ocean Test of Concrete Spherical Structures", Technical Report R-805. Civil Engineering Laboratory, Port Hueneme, Calif., March 1974.

44. Haynes, Harvey H., "Research and Development of Deep Submersible Concrete Structures", Proceedings of FIP Symposium at Tbilisi, 1972. London: 1973, pp. 180-185.

45. Wallace, George B., "Joints and Cracks in Concrete Water-Holding Structures", Symposium on Concrete in Aqueous Environments, ACI Publication SP-8, Detroit: 1964, pp. 21-42.

APPENDIX A: COST DATA FOR OTEC ALTERNATIVES[23]
(in millions of dollars)

## CARNEGIE-MELLON UNIVERSITY DESIGN

| Baseline Description (100 $Mw_e$ net) | Unit Cost |
|---|---|
| Electrical (at bus bar) | 7.44 |
| Heat exchanger | 56.96 |
| Cold water pumps | 7.63 |
| Warm water pumps | 8.81 |
| Turbine generators | 3.62 |
| Ammonia piping, etc. | 5.22 |
| Ammonia pumps | .42 |
| Ammonia inventory | 1.01 |
| Controls power modules | 1.71 |
| Auxiliary system power modules | 5.55 |
| Misc. power | -- |
| Shell struct. power modules | -- |
| Superstructure misc. | -- |
| Platform structure | 57.0 |
| Mooring system | 7.50 |
| Engineering and field start | 18.51 |
| Deployment | .96 |
| Home office services | 3.56 |
| Total | 165.45 |

Capital Cost in \$/kw = \$1655.

UNIVERSITY OF MASSACHUSETTS DESIGN

| Baseline Description (400 $Mw_e$ net) | Unit Cost |
|---|---|
| Electrical (at busbar) | 27.49 |
| Heat exchanger (6) | 291.83 |
| Cold water pumps (8) | 18.44 |
| Warm water pumps (8) | 27.12 |
| Turbine generators (8) | 20.57 |
| Propane piping, etc. | 33.10 |
| Propane pumps | 9.76 |
| Propane inventory | 3.02 |
| Controls power modules | 2.88 |
| Auxiliary system power modules | 8.06 |
| Misc. power | -- |
| Shell struct. power modules | -- |
| Superstructure misc. | -- |
| Platform structure | 82.76 |
| Mooring system | 37.0 |
| Engineering and field start | 72.68 |
| Deployment | .24 |
| Home office services | 13.98 |
| Total | 646.01 |

Capital cost in $/kw = $1615

ANDERSON'S DESIGN

| Baseline Description (100 $Mw_e$ net) | Unit Cost |
|---|---|
| Electrical (at busbar) | 9.01 |
| Heat exchanger | 32.47 |
| Cold water pumps | 4.75 |
| Warm water pumps | 7.55 |
| Turbine generators | 6.86 |
| R-12/31 piping, etc. | 12.19 |
| R-12/31 pumps | -- |
| R-12/31 inventory | 6.55 |
| Controls power modules | 1.18 |
| Auxiliary system power modules | 5.53 |
| Misc. power | -- |
| Shell struct. power modules | -- |
| Superstructure misc. | -- |
| Platform structure | 30.30 |
| Mooring system (dynamic) | 3.38 |
| Engineering and field start | 18.15 |
| Deployment | .66 |
| Home office services | 3.02 |
| Total | 142.81 |

Capital cost in $/kw = $1428

## TRW BASELINE

| Baseline Description (100 $Mw_e$ net) | Unit Cost |
|---|---|
| Electrical (at busbar) | 7.44 |
| Heat exchanger (8) | 79.83 |
| Cold water pumps (4) | 3.73 |
| Warm water pumps (4) | 2.83 |
| Turbine Generators (4) | 3.65 |
| Ammonia piping, etc. | 5.22 |
| Ammonia pumps | .42 |
| Ammonia inventory | 1.01 |
| Controls power modules | 1.18 |
| Auxiliary system power modules | 5.55 |
| Misc. power | -- |
| Shell struct. power modules | -- |
| Superstructure misc. | -- |
| Platform structure | 68.79 |
| Mooring system (dynamic) | 1.50 |
| Engineering and field start | 23.55 |
| Deployment | .96 |
| Home office services | 4.53 |
| Total | 210.19 |

Capital cost in $/kw = $2102

## LOCKHEED BASELINE [22]

| Baseline Description (160 $Mw_e$ net) | Unit Cost |
|---|---|
| Electrical (at busbar) | 9.2 |
| Heat exchanger (8) | 236.6 |
| Cold water pumps (16) | 14.1 |
| Warm water pumps (16) | 11.9 |
| Turbine generators (8) | 12.4 |
| Ammonia piping, etc. | 4.7 |
| Ammonia pumps | .7 |
| Ammonia inventory | .3 |
| Controls power modules | .2 |
| Auxiliary system power modules | 6.1 |
| Misc. power | 8.0 |
| Shell struct. power modules | 27.6 |
| Superstructure misc. | 4.3 |
| Platform structure | 42.9 |
| Mooring system | 31.0 |
| Engineering and field start | 5.3 |
| Deployment | 5.0 |
| Home office services | -- |
| Total | 420.3 |

Capital cost in \$/kw = \$2621

Thesis 165470
K2595 Keller
An analysis of ocean thermal energy conversion systems.